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VIBRATION TESTING OF RESILIENT PACKAGE CUSHIONING MATERIALS

GEORGE ZELL

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PICATINNY ARSENAL
DOVER, NEW JERSEY

VIBRATION TESTING OF RESILIENT PACKAGE CUSHIONING MATERIALS

by

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Feltman Research Laboratories
Picatinny Arsenal
Dover, N. J.

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OBJECTIVES

1. To establish performance parameters for evaluating the vibratory response of resilient package cushioning materials.

2. To develop apparatus capable of defining vibratory response as a function of applicable variables.

3. To begin testing of specific cushioning materials.

SUMMARY

Performance criteria for cushioning materials and tentative test methods have been evolved through coordination with an ASTM task group from government and industry. Testing equipment which meets tentative ASTM requirements has been developed. Preliminary tests have been conducted on a resilient expanded polystyrene foam (in 3 densities) and a polyether urethane foam (in one density).

When vibrated under static loads known to provide optimum shock mitigation (0.6 to 1.5 psi) the polystyrene foam experienced pronounced permanent set (up to 36% of original thickness) after partial completion of the vibration transmissibility test schedule.

The set induced in similar new cushions by vibration generally exceeded the set induced (under the same static loads) by 5 successive 36-inch free-fall impacts.

The 5-impact sequence gave the vibrated polystyrene cushions permanent sets up to 78%. Transmitted accelerations were up to 220% of those for unvibrated cushions.

Polyurethane foam subjected to the entire vibration transmissibility schedule at maximum recommended loading (0.16 psi) experienced negligible permanent set. Complete impact test data is not available for this foam; however, stress-strain curves taken after vibration indicated negligible change. Vibration amplification at resonance averaged 6 for polystyrene foam and 3 for polyurethane foam.

CONCLUSIONS

Since the in-transit environment of critical fragile items is characterized by vibratory loads as well as impulsive ones, the transmission characteristics and fatigue susceptibility of cushioning materials should be defined for vibration as well as for impact.

Preliminary testing indicates that certain materials with favorable shock mitigating properties in specific load ranges may be extremely susceptible to vibratory fatigue at those loadings.

Progressive degeneration of cushioning material during vibration transmissibility testing impairs the validity of transmissibility data. The techniques used should effect the most rapid completion of transmissibility tests, while assuring the attainment of steady state response at all frequencies. This objective can be most consistently attained by using automatically programmed frequency sweeps.

RECOMMENDATIONS

Vibration transmissibility and fatigue testing should be continued on materials whose properties are otherwise promising for military application.

More extensive tests of polyurethane foam should be conducted, in view of its favorable response in initial tests at room temperature.

Because of their greater apparent resistance to vibration fatigue damage, resilient polystyrene foams of higher density should be used in packaging in preference to those of lower density. The optimum shock loading ranges and efficiencies of this material vary little with density.

Resilient polystyrene foam should not be used at static loads in excess of 1.0 psi because of its apparent susceptibility to fatigue when loaded above this level.

Vibration testing of resilient polystyrene foam should be continued at loads under 0.6 psi.

Vibration testing should be initiated on other materials with favorable shock mitigating qualities at static loads in excess of 1.0 psi to determine their adequacy as replacements for polystyrene foam in high stress applications.

Sweep frequency testing should be initiated to reduce transmissibility test time associated with discrete frequency testing.

INTRODUCTION

Rational design of cushioning systems for package shock mitigation has been facilitated by the increasing availability of static stress vs acceleration data for specific cushioning materials. The need for similar information on the vibratory behavior of cushion materials has been generally recognized by military agencies and industry groups specifically responsible for the safe shipment of complex, fragile equipment.

Such information is generally available on prefabricated shock mountings manufactured by reputable companies specializing in shock and vibration control. In contrast, the production of bulk cushioning materials is, in most instances, a sideline for the manufacturer. Cushioning may be a byproduct or a peripheral application of a basic product. Thus most cushion material manufacturers are unable to furnish vibration data on their materials and are reluctant to make the investment in technology and equipment needed for obtaining such data.

However, the serious package designer encounters many isolation problems to which the application of distributed cushion material offers the most attractive solution. Faced with a total lack of pertinent cushion vibration data for high priority development programs, many military agencies and contractors have conducted extensive test programs keyed to their particular applications and requirements. Because such tests have been intimately involved with the development of a particular item or subsystem, the results are not readily useful to designers in other areas. Where testing is performed by a private corporation for its own internal purposes, there is an understandable reluctance to release information to the possible advantage of competitors. Consequently, a designer seeking vibration data on an otherwise promising cushion material is faced with a potentially fruitless search; or, having located a suitable material, he finds it too limited in scope for his particular application. As a result, the careful designer often considers only materials he has personally investigated, rather than risk serious problems by using untested materials.

To focus attention on the problem area and establish cushion vibration test criteria worthy of general recognition, engineers of the Boeing Company fostered the formation of an ASTM task force. Acting in accord with the trend towards military recognition of suitable commercial standards, Picatinny Arsenal, having previously initiated a similar program, assumed an active role in the task group, together with other government and industry representatives. The interest of the ASTM D-10 committee (Interior Packaging Subcommittee) in the definition of the shock mitigating qualities of cushioning materials had previously resulted in the tentative D-1596-59T test specification. This test standard has received considerable military and industrial recognition and has greatly rationalized the use of cushion materials in mitigating shock. In the definition of "shock response," a simple, direct test procedure (the drop test) was available. Does a similarly simple approach present itself in the area of cushion vibration response? Thoughtful consideration of all aspects of the problem precludes an optimistic answer.

The difference stems from the nature of the phenomena under investigation. Shock is a discontinuous function in time (in the case of the cushion drop test, an abrupt velocity change) that induces transient vibration of the cushion system and cushioned item at their own natural frequencies. Thus, the simple validity of the ASTM cushion shock test lies in its straightforward generation of a velocity shock input to the cushion by means of a drop test. Vibration, on the other hand, is a continuously varying function that induces continuous oscillation of the cushion and cushioned item at the frequencies comprising the forcing function. The reproduction or duplication of these vibratory inputs in a laboratory is extremely costly, quite complex, and requires sophisticated equipment, as the following considerations indicate:

The vibrations encountered in transit span a wide range of frequencies from the 2.5 cps motions of railroad vibration (Ref 1) to 300 cps in aircraft cargo beds (Ref 2). The amplitudes associated with these vibrations range from 3.5 inches peak-to-peak for the former to .001 inch for the latter. The peak vibratory acceleration documented for railroad vibration (Ref 1) is 1.3 g, while the value recorded for cargo aircraft (Ref 2) approaches 5 g. It is evident, therefore, that relatively sophisticated vibration apparatus is required to simulate this wide range of vibrations in the laboratory.

Transportation vibration varies greatly in character as well as frequency. Railroad vibration has specific periodic content, as does the vibration induced in propeller-driven aircraft. Vibration in wheeled and tracked vehicles is, however, predominantly nonperiodic and thus can be adequately described only in terms of a combination of spectral and statistical analyses (Ref 3). The primary forcing functions stem from irregularities in the surface being traversed. The vibration induced in the vehicles is a complex of shock-excited transient vibration and forced vibration in response to continuous (but not necessarily periodic) forcing functions. Aberdeen Proving Ground has recently published reports documenting the amplitude distributions and frequency spectra of vibrations induced in a number of military transport vehicles by operation over various road courses at APG (Ref 4). While these analyses are generally indicative of a random, nonperiodic, quasi-gaussian vibration, APG has formulated an "equivalent sinusoidal" spectrum for test with conventional sinusoidal vibration equipment. The peak sinusoidal acceleration specified is 1.6 g for tracked vehicles and 0.8 g for wheeled vehicles.

An overall sinusoidal vibration envelope had been tentatively adopted by ASTM before the release of the Aberdeen data. This spectrum (Fig 10, p 40) originally appeared in Harris' & Crede's compilation, "The Shock and Vibration Handbook," and was subsequently adopted by the Air Force in MIL-STD-810 as representative of the transportation vibration environment. The good agreement of the ASTM envelope with the APG curve for tracked vehicles is evident (Fig 10A, p 41).

It should not be inferred that the ASTM envelope represents the exact vibration environment to which a specific cushioned package will be subject in transit. Rather it should be regarded as defining the limits of magnitude of vibration that a package may encounter as a continuous periodic disturbance while being carried by any of the common modes of transportation. Response at reduced levels of excitation is studied because of the nonlinear character of cushion vibration response. (The limitation of the envelope to 1.0-inch double amplitude at the low frequency end of the spectrum represents an acknowledgement of the limits of electrodynamic vibrator capabilities.)

Throughout this continuing investigation, the ASTM spectrum will be used to define the following two categories of cushion vibration response:

1. The vibration transmissibility function for steady-state sinusoidal excitation as determined by static loading, cushion thickness, temperature, vibration amplitude, and frequency. In this work, the extent to which

particular cushion configurations transmit environmental vibration to the cushioned item will be considered.

2. Comparison of fatigue susceptibilities of cushioning materials under continuous vibration, as a function of static loading, cushion thickness, and temperature. The ASTM envelope will be swept repetitively for an hour as a test criterion. Indices used to determine cushion fatigue will be: thickness loss, change in the static stress-strain curve, change in impact transmission, and impact set.

The materials were tested in a simulated package configuration, essentially a single-degree-of-freedom system with variable static loading capabilities (Fig 1, p 33). The configuration is designed to allow such vibration responses as derive from the intrinsic qualities of the material under test, but suppress all other responses in the simulated package and cushioned mass. Details of the test assembly are shown in Figures 1 through 3 (pp 33 through 35). Underlying theoretical and practical considerations affecting the test apparatus and test procedures are detailed in subsequent sections of this report. Figure 4 (p 36) shows the test assembly mounted on an electro-hydraulic vibrator. This vibrator was used for most of the sine wave testing. Because of its long stroke capabilities at low frequencies and its programmability, the possibility also exists of using this vibrator to investigate the response of cushioning systems to low frequency Gaussian vibration as encountered in vehicular transport. This area will be studied later in the program.

The preliminary transmissibility testing summarized earlier was performed at discrete frequencies. Although this approach is valid from a theoretical point of view, the time required for comprehensive determination of response spectra was found prohibitive. It is felt that the progressive compaction experienced by the polystyrene foam during the lengthy data acquisition procedure impaired the validity of the resultant response data. To reduce and standardize transmissibility test time, subsequent testing will utilize automatically programmed frequency sweeps while simultaneously recording response data.

BACKGROUND THEORY

A relatively soft material (cushioning) is often inserted between a fragile item and its exterior shipping container to reduce the transmission of shock loads to the item. Such materials tend also to drastically alter the vibration transmitting qualities of the package. If protection against multiple shocks is desired (as is usually the case), the cushioning must have good recoverability or "resilience." When a fragile item is supported in such cushioning, a mechanical system with inherently oscillatory properties is created. Such systems tend to be extremely frequency discriminatory in their transmission of steady-state vibrations. The first concern of this report is the development of methods for the experimental definition of this phenomenon, subject to variation of response-determining factors that are variable in actual cushion application. A second concern is the measurement of cushion degradation under vibratory loading.

The development of sound test criteria in this area is made easier if the investigator has a general insight into the behavior of the cushioned package under vibration. This insight may be gained by studying the theoretical response of a simplified model. The influence of the simplifying assumptions may then be considered, to more closely approximate the response of typical cushion systems.

A schematic view of the most common cushion application is shown in Figure 11 (p 42). A fragile item is supported within an exterior container by cushion pads on all its faces. The type and thickness of the pads are normally determined by shock mitigating requirements. Cushions on opposite faces of the item are usually identical if, as this report assumes, the bearing areas are equal. Motion (vibration) in the vertical direction only is considered. It is assumed that the side cushions limit the motion of the cushioned item to the vertical direction while introducing a minimum of frictional drag. The weight of the cushion is assumed to be negligible in comparison to that of the cushioned item, and the container and item are assumed to be extremely rigid in comparison to the cushions. Subject to these simplifications, the cushioned package is strongly analogous to the classical single-degree-of-freedom system depicted in Figure 12 (p 42).

In this model, a rigid mass, M, is attached to a rigid base structure via a massless spring. The spring is linear, i.e., it obeys Hooke's law in tension and compression ($F = -KX$). K is the spring constant in pounds/inch. X is the relative compression or extension of the spring at any instant in time, t, and is equal to the instantaneous distance between the mass and the base. $X = A_m - A_b$, where A_m and A_b are the instantaneous displacements from their rest positions of the mass and the base, respectively. The mass is constrained to move in the vertical direction with no restraint in that direction other than instantaneous spring force.

In the model, the mass represents the cushioned item, the rigid base the container, and the spring the upper and lower cushions. Compressive forces in the spring reflect compressive forces in the lower cushion, while compressive forces in the upper cushion appear as tensile forces in the spring. While actual cushions display varying degrees of departure from linearity, the compressive force vs displacement relationship of the cushions will be initially assumed to be linear, i.e., analogous with the model. The vibration response of the model will now be considered.

Vibration was broadly defined in the introduction as the continuous variation of motion about a reference point. While the variation in time may in some instances be totally random, the motions generally associated with the term are recurrent or periodic in time. Of these, the most fundamental in concept is sinusoidal vibration, or simple harmonic motion. The nature and physical effect of more complex vibrations may often be defined (analytically and experimentally) in terms of sinusoidal parameters. A sinusoidal displacement function is plotted against time in Figure 13 (p 43). The salient features of the vibration are apparent: The displacement from the rest position, A_B, reaches equal positive (upward) and negative (downward) positions in time, and the motion repeats itself in a time interval, T, defined as the period of the vibration. The reciprocal of the period is the frequency, $f = 1/T$, the number of complete repetitions occurring per unit time, usually in cycles per second, (cps). The quantity N is a constant, $N = 2\pi$. It is desired to establish the motion of the mass, M (the cushioned item), in the simplified model in response to the sinusoidal vibration of Figure 13 applied to the "base" (the exterior container).

Intuitively, we know that the response of the mass to vibration of the base will also be vibratory. This may be verified by solving the differential equation of mass motion. Referring to Figure 12 (p 42) and using Newton's second law of motion, we find that the instantaneous mass acceleration, $\ddot{A}_m$,[1] is equal to the instantaneous spring force divided by the mass magnitude, M.

$$\ddot{A}_m = -KX/M = (-K/M)(A_m - A_o \sin Nft).$$

[1]The dots superimposed over functions variable in time symbolize the time derivatives of the functions, the number of dots indicating the order of the derivative. Since A_m is the instantaneous mass displacement, $\ddot{A}_m$ is the mass acceleration.

The steady-state (long-term) solution found in the literature is

$$A_m = A_o \times [1/(1 - f^2/f_n^2)] \times \sin (Nft - \phi_f).$$

The last term of the solution indicates that the mass motion is also a sinusoidal vibration of the same frequency as the base disturbance. A frequency-dependent time displacement of the response vibration is indicated by the quantity, ϕ_f. The prime point of concern is the magnitude of the mass vibration. The maximum amplitude of the vibratory mass response, A_r, is seen to depend not only on the magnitude of the base vibration, but on its frequency, f, as well. The response at a particular frequency and amplitude of excitation is determined by the quantity, f_n, which is the natural frequency of vibration of the particular spring-mass system, i.e., the frequency at which the mass will vibrate in response to transient as opposed to periodic disturbances. Its value may be determined analytically by assuming a momentary displacement of the mass relative to the base and solving the pertinent equation for mass motion after release: $\ddot{A}_m = -K/M \times A_m$. Free oscillation of the mass is found to take place at a frequency, $f_n = 1/N \times (K/M)^{1/2}$.

The ratio of the maximum vibratory amplitude of the mass, A_r, to that of the base, A_o, was shown to be equal to $1/(1 - f^2/f_n^2)$. This ratio is plotted in Figure 14 (p 44) against the frequency ratio f/f_n. The response ratio is commonly termed the "transmissibility" of the spring-mass (cushioned) system. In Figure 14, the extreme variation of vibration transmission with frequency, postulated earlier, is evident. The exciting (base) vibration is seen to be transmitted unaltered in intensity (transmissibility = 1) only at frequencies well below the natural frequency, f_n. At frequencies numerically near f_n, the mass vibration is observed to attain extremely large values (theoretically infinite at $f/f_n = 1$). At exciting frequencies beyond $\sqrt{2}\, f_n$, the vibration transmitted to the mass (the cushioned item) decreases rapidly with frequency (transmissibility less than 1). The external vibration is then said to be attenuated and the cushioned mass is said to be isolated from the higher frequency vibrations.

This isolation of the cushioned item from external vibration is an obviously favorable aspect of the resilient cushion suspension. The intensification of vibration at frequencies near the natural frequency is, however, an unavoidable corollary of the isolation phenomena. The natural frequency was noted to be determined by the ratio K/M, K being the spring constant of the cushioning and M the mass of the cushioned item. It would appear that this frequency could be lowered below any disturbing vibrations by making the "stiffness" of the cushion sufficiently low. This practice is feasible in the design of vibration isolators for equipment installed in aircraft, since the exciting vibrations are relatively high in frequency. However, the vibrations encountered by the cushioned package in shipment by rail and road vehicles was shown to extend down to 2.5 cps. To ensure that no amplification of vibration above this frequency would occur, a natural frequency for the cushioned package of 1.8 cps would be required ($2.5/\sqrt{2}$). From energy considerations, it can be demonstrated that the natural frequency of a linear cushion that would transmit a peak acceleration of 10 g's in a 3-foot free fall would be 3.7 cps. The thickness of cushion required for the latter application would be 10 inches, if an overall cushion efficiency of 36% is assumed. (The assumed efficiency is relatively high for actual cushions with approximately linear stiffness.) The thickness required for the 1.8-cps system to preclude its bottoming out in a 3-foot fall would be of the order of 20 inches (based on a 36% efficiency). From these considerations, it must be concluded that:

1. The natural frequencies of cushioned packages designed to meet most shock mitigation requirements will fall in frequency ranges where they may be excited by rail or vehicular vibration.

2. Designing a cushion system whose natural frequency falls below all vibrational frequencies that may be encountered in transit is impractical.

These conclusions make the excitation of the natural frequency of cushioned packages a general problem of the packaging engineer.

Theoretically, the response of the cushioned mass when excited at the natural frequency of the cushion suspension would be "infinite." Obviously, the vibration of the cushioned mass cannot exceed the confines of the external container. Limiting influences other than container size are operative, however.

Many of the newer cushioning materials are foamed elastomeric polymers (polyurethane, polyvinyl chloride, polyethylene, etc.) These materials in a homogeneous state generally manifest viscoelastic properties (i.e., their resistance to deformation is rate sensitive). This characteristic may be expected to extend to their foamed state as well. The simplest rheological model for viscoelastic behavior is a linear spring shunted by a linear dashpot. Such a dashpot may be considered as connected across the linear spring of the single-degree-of-freedom model of the cushioned package (Fig 12, p 42) and its effect on vibration transmissibility may then be examined.

The generalized equation of mass motion becomes:

$$\ddot{A}_m = -(KX/M + C\dot{X}/M)$$

where $\dot{X}$ is the differential velocity of the mass and the base, and C is the coefficient of viscous damping. The steady-state solution for mass displacement with the sinusoidal vibration of Figure 13 (p 43) again applied to the base is found to be

$$A_m = A_o \times \left[\frac{1 + (2rf/f_n)^2}{(1 - f^2/f_n^2)^2 + (2rf/f_n)^2}\right]^{1/2} \times \sin(Nft - \phi_{f,r})$$

(The phase angle between excitation and response vibration, $\phi_{f,r}$, varies with the frequency f and the damping ratio r—hence the subscripts. Again the magnitude of the vibration is of primary interest). The maximum vibratory displacement value for the mass is given by the first two terms of the solution. The response ratio or "transmissibility" is the second term. Transmissibility is plotted against frequency in Figure 15 (p 45) with r as a parameter. The quantity r is defined as the "damping ratio." $r = C/C_c$, where C_c is that coefficient of viscous damping that is just large enough to suppress oscillation of the cushion system in response to transient excitations.

Examination of Figure 15 indicates that increased viscous damping of the cushion reduces the magnification at frequencies in the vicinity of the natural frequency at the expense of reducing the attenuation or isolation of the higher frequencies of excitation. With increased damping, the frequency of maximum response is observed to be shifted slightly downward from the natural frequency. The frequency of maximum transmissibility is defined as the "resonant frequency." Since viscous damping is inherent in actual cushioning materials, the undamped natural frequency for such materials cannot be determined by transmissibility testing, and the resonant frequency and the amplification ratio associated with it become of primary concern.

"Damping" is the general term applied to any mechanism that results in the dissipation of energy from vibrating systems. "Pure" viscous damping

(damping force dependent on the first power of velocity) is but one of many damping relationships that may be encountered—often within the same cushioning material. The damping force may be proportional to higher powers of the relative velocity or may be relatively independent of velocity. The latter type of damping is generally referred to as "structural" or "hysteresis" damping. The stress-strain curves for the two materials used in the preliminary testing summarized earlier appear in Figures 16 and 17 (pp 46 and 47). These curves were taken at a loading rate of 10 inches per minute and, relative to loading rates in vibration, they may be considered as "static" curves. The deviation between the loading and unloading curves is evident for both materials. The area between these curves represents the energy lost per cubic inch of material during a half-cycle of vibration, independent of oscillation rate. This hysteresis-type loss is generally associated with nonlinearity in the stress-strain relationship and, hence, in the "stiffness" of the cushion. The spring constant, K, of a particular cushion is directly proportional to its loaded area and inversely proportional to its thickness. The slope of the stress-strain curve, therefore, represents the rationalized, or "per-unit" spring constant. The nonlinearity of stiffness in the two materials is obvious in Figures 16 and 17. On brief reflection, it is apparent that every cushion loaded in compression must eventually evidence nonlinearity as the cushion bottoms out.

The two general types of nonlinearity in cushion stiffness are:

1. Stiffening nonlinearity, wherein the slope of the stress-strain curve increases with increased strain.

2. Softening nonlinearity, where the slope of the stress-strain curve manifests a decrease with increased strain.

The changes in slope displayed may be relatively gradual or quite abrupt. The stiffening type of nonlinearity is generally associated with "bottoming" of the fibers or cells of the cushion and hence is the most commonly encountered form. (Stiffening of the polystyrene foam in Figure 16 is seen to take place at 35-40% strain.) The softening phenomenon is usually associated with a "buckling" mechanism in the cushion structure. Cushions which display a softening in a particular range of strain must eventually undergo a stiffening with further increase of strain as bottoming ensues. This dual nonlinearity is very apparent in the stress-strain curve of the polyurethane (Fig 17).

Previous analyses of the vibration transmissibility characteristics of the cushioned package have assumed linearity of stiffness and damping. On the basis of these assumptions, a vibration response ratio (transmissibility) dependent only on the frequency was derived. If these assumptions of linearity were applicable to actual cushioning materials, the experimental determination of their response would be necessary at only a single level of excitation for each frequency. The specific level used would not have to reflect the actual level encountered in the environment at each frequency. In view of the generally nonlinear properties of cushioning media, are these test-simplifying considerations also valid in the response determination of nonlinear systems?

The effects of nonlinear stiffness on the vibration response spectrum of cushioning can be most easily comprehended by considering its effects on the determinant factors in response revealed by the linear model. The response of the cushioned mass to a continuous external vibration was shown to be greatest when the forcing frequency coincided with the frequency of free oscillation of the cushioned mass (the natural frequency). This frequency was indicated to be determined by the ratio of the spring constant, K, to the mass of the cushioned item.

$$(f_n = 1/2\pi \, (K/M)^{1/2})$$

For a nonlinear cushion, the spring constant varies with displacement. The effective natural frequency must also vary with displacement. Consider a stiffening cushioning system. If the cushioned mass is depressed into the displacement range where stiffening occurs, and then released, the mass will oscillate between the upper and lower cushions. The initial oscillation will take place in a relatively short time interval because of the high average stiffness operable over the cycle. Because of damping in the cushion, the mass will not reach the initial point of release on completion of the first cycle. In subsequent cycles of free oscillation, the average stiffness will become progressively less and the time required to complete successive cycles will grow correspondingly longer. The "frequency" of free vibration ($f_n = 1/T_n$) is thus dependent on the range of displacement over which the oscillation is instantaneously occurring. Now consider the effect of this phenomenon on the response to forced vibration.

Assume that a stiffening system is excited by a relatively low level of vibration at its resonant frequency as determined by an initially linear stress-strain curve. The maximum amplitude of mass motion will be limited by the damping effect of the cushion. The mass response will be at a maximum for the particular level of excitation. If the excitation level is then increased (maintaining the same frequency) until the mass is driven into the stiffening region of the cushion, the response is no longer maximal for the new level of excitation, since the effective natural frequency has been shifted to a higher frequency by the increase in the average spring constant. Maintaining the higher level of excitation while increasing the frequency will result in a maximum response being achieved at a higher frequency, the resonant frequency for the higher level of excitation. These effects are illustrated in Figure 18 (p 48) in characteristic response curves for a stiffening cushion system. The salient divergences from linear system response are apparent. They may be summarized as follows:

1. No single resonant frequency exists independent of excitation amplitude.

2. Transmissibility at resonance varies with the level of excitation. A single transmissibility curve does not adequately portray all response characteristics.

3. In some frequency ranges, more than one level of response is possible at a specific frequency and input level. In Figure 18, at a frequency of excitation f', and level of excitation 4A, three possible levels of response are predicted from the intercepts of the line f = f' with the pertinent response curve. Experimentally, the upper level of response will be achieved if the frequency f' is approached from a lower frequency, and the lowest value by approaching f' from a higher frequency. The response condition represented by the intermediate intercept is physically unstable and difficult to induce experimentally.

The response phenomena described for stiffening cushions occur in softening systems as well. In these systems, however, the average stiffness decreases with amplitude and thus the resonant frequency decreases with excitation amplitude (Fig 19, p 49). As has been noted, however, "softening" cushioning systems must ultimately manifest a "stiffening" as bottoming is approached. As a result, under sufficiently intense excitation, the f_n trajectory for such systems may be expected to curve back to the right, i. e., toward higher frequencies.

In the preceding discussions, an attempt has been made to predict the nature of the characteristic vibration response spectra of mass-loaded resilient cushioning. The intent of these discussions has been threefold:

1. To establish a community of understanding and emphasis with those who may in the future wish to apply response data derived from this program.

2. To relate the character of the response curves and pertinent parameters to the test equipment and procedures required for their reliable experimental definition.

3. To rationalize and justify the complexity of the test procedure and equipment requirements as compared to the physical simplicity of the cushion shock test.

The degree to which aims 1 and 3 have been realized cannot be gauged. However, the implications of the discussions in relation to experimental testing may be summarized as follows:

1. While factors such as container and cushioned item resilience, rotational vibration modes, and frictional damping introduced by side cushioning affect the vibration induced in the cushioned item during actual shipment, they are extraneous to evaluation of intrinsic vibratory qualities of cushioning and should be suppressed in the test package. Thus the container utilized as a test vehicle should be rigid, as should the cushioned item (test mass). (Definition: rigid = nonresonant in the frequence range used in the testing.) Rocking modes of the test mass must be suppressed, i. e., the test mass must be guided, since the force center of cushion samples may not be centrally located. The restraint of the guides must be "frictionless," since it is desired that vibration response be limited only by properties of the cushion suspension. The test package should constitute, in brief, a nonlinear single-degree-of-freedom system.

2. It was shown that the ratio of cushion stiffness to cushioned item mass, K/M, is a strongly determinant factor in the vibration response of cushion systems. The stiffness of a cushion = $K = EA/T_c$ where E is the modulus of the cushion (slope of the stress-strain curve), A is the cushion area, and T_c the cushion thickness. The mass is the weight of the cushioned item divided by the gravitational acceleration, g: $M = W/g$. Therefore: $K/M = Eg/T_c \times 1/(W/A)$. E and T_c are determined for a particular thickness

of a particular type of material. The variable response factor is then the quantity W/A, the static stress imposed on the bottom cushion by the test mass, in psi. References to drop test data for unvibrated cushions in Figures 23 through 25 (pp 53 through 55) indicate that this parameter also determines the peak shock response acceleration for a given material, drop height, and cushion thickness. Since it is unlikely that a cushioning material will be used in a loading range where it is ineffective as a shock isolator, static stresses for vibration testing should encompass the "optimum range" for shock mitigation–i. e., the loading range in the pertinent acceleration-static stress curve wherein acceleration is at a minimum. However, the fixed parameter the cushion designer encounters is the static stress imposed by the specific item to be cushioned. Therefore the overall range of static stresses that can be imposed in the test package should span the range of static stresses encountered in equipment to be cushioned. As the static stress is a ratio (W/A), its value for a particular test could be achieved by varying either W or A. Of the two routes, the variation of weight is more satisfactory since a uniform size of test sample is desired. The attainment of high stress by decreasing specimen area tends to make the test results more susceptible to material variations.

3. The frequency spectrum of in-transit vibration extends from 2.5 to 300 cps. Since the response of cushion systems varies drastically with frequency, the total experimental evaluation of cushion vibration response requires the excitation of the complete environmental frequency range in the laboratory. The vibration response ratio of cushioning systems is generally dependent on amplitude- as well as frequency. Therefore, thorough evaluation of their response requires the excitation of the systems at several levels of amplitude, up to the maximum levels that occur in the transportation environment.

4. The instrumentation used to measure the vibration response must be accurate over the range of frequencies and amplitudes encountered.

DISCUSSION OF TEST APPARATUS AND PROCEDURES

The hardware developed at Picatinny to satisfy the requirement for a single-degree-of-freedom test fixture is shown in Figures 1 through 4. The maximum cushion area that can be loaded is 100 square inches (10 in. × 10 in.). With this load bearing area, the static stress range that can be achieved is .06 to 2.00 psi. The bearing stress can be proportionately increased by decreasing the cushion area. The loading of the cushion by the test mass is over the total area in all cases. With maximum test load (200 lb) the maximum cushion thickness accepted is 6 inches (for a pair of samples, upper and lower). Greater thicknesses can be tested at reduced total loads. The entire test assembly (hardware) has been tested for resonance with accelerometer and stroboscopic instrumentation in the 200-300 cps frequency range. All resonances revealed have been eliminated. The lower limit of static stress attainable with a full 100-square-inch bearing area (.06 psi) is obtained with a 12" × 12" × 5" aluminum honeycomb "sandwich" bonded with epoxy. The sandwich is equipped with 2 teflon "shoes" at each corner (1 upper and 1 lower) to provide the required guidance within the "guide fixture" (exterior container). The "sandwich" or guide weight is included in all test loads to insure single-degree-of-freedom motion. Auxiliary weights, used as required, are bolted to the guide weight to form a rigid, nonresonant "cushioned mass" that is supported between the upper and lower test cushions. Test cushions are fastened to the top plate and guide fixture with double-backed tape. This prevents the cushions from moving relative to the exterior container when the container acceleration exceeds 1 g. An electronically programmable hydraulic vibrator is used to excite the test package. The overall frequency capability is dc to 400 cps. The vibrator is capable of attaining the peak sinusoidal accelerations specified by the ASTM envelope, which is defined partly in terms of constant vibratory displacement and partly in terms of constant vibratory acceleration. The response tests are conducted at constant acceleration levels that "blanket" the envelope. The "motion response" vibration that has been referred to in the discussions is the absolute vibration of the mass, i. e., the motion with respect to an external inertial reference. However, the absolute mass response can be most readily measured with an accelerometer, since the output of this type of transducer is inherently proportional to the absolute (inertially referred) mass acceleration. The measurement of response in terms of acceleration also furnishes the response data in a form more readily translated into stress.

The comprehensive data obtained during this investigation will be families of Acceleration-Response-vs-Frequency curves for specific sinusoidal accelerations, similar to Figures 18 and 19. Each family of curves will be pertinent to a particular type of cushioning material of specific thickness and under specific static loading. Since each of four thicknesses will be tested at up to five levels of static stress, the data package completely describing the vibration response of a material may consist of up to 20 families of curves. This volume of data would tend to awe the potential user.

In order to "capsulize" each material's vibration response, a cross-plot of its resonance characteristics as a function of the governing variables can be constructed. Such a plot is shown in Figure 22 (p 52). Since it has been shown that, in general, no cushioned package can feasibly be designed so that there is no possibility of excitation of its resonant frequency during shipment, it is assumed that the resonant response of a cushion system will constitute a strong initial criterion for its evaluation. It is further assumed that the cushion designer has already determined that the material has satisfactory shock mitigating properties. The static stress is predetermined by the item to be cushioned. On the basis of item fragility, a particular thickness of the material has been found necessary from static stress vs acceleration curves.

To determine the resonant response characteristics, the designer selects the cross-plot for the required thickness, locates the static stress imposed by the cushioned item, and follows the curve for that stress to its intersection with the curve for the expected level of excitation in g's. Horizontal projection of the intersection to the "Resonance Response Amplitude" axis establishes the maximum response acceleration. Similarly, vertical projection establishes the resonant frequency. For example, in Figure 22: The response acceleration of cushion A, thickness T_c, is desired, with an imposed static stress of S_4 and an excitation acceleration of G_1. A maximum response C_r is found to occur at frequency f_r. The dotted response curves indicate how the resonant response grid is constructed.

The accelerometers used in the testing were ceramic piezoelectric types. These accelerometers were chosen because of their small size and attendant ease of mounting. A second consideration was their tolerance of the +160°F and −65°F temperatures to be employed in future testing. Being capacitive devices, their low frequency cutoff is fixed by the RC time constant, determined by the total capacitance of the accelerometer and connecting cables, and the input impedance of the electronics. The acceleration analog voltage from the accelerometer is coupled to response-indicating or recording devices via an electrometer amplifier (dc) or a cathode follower to insure a flat response extending below the lowest frequency generated in testing. The high resonant frequency of the accelerometer (30 kc) makes the upper limit of response a function of the frequency limitation imposed by subsequent elements of the indicating or recording system.

The interim system used to measure the acceleration response in the preliminary testing is shown in Figure 5. The vibration was manually programmed to excite specific peak sinusoidal accelerations and frequencies. The response accelerations were monitored on the oscilloscope and recorded manually. The resultant data points, when plotted, were to be connected to form continuous response vs frequency plots for the specific levels of exciting acceleration generated. A family of response curves, like the one shown in Figure 18, was the desired end-product. The rapid change of response with frequency in the resonant frequency range necessitated a large number of data points to adequately define the resonant peak. As noted in the Introduction, a totally unexpected degree of cushion deterioration was experienced during the data acquisition interval. Several disturbing philosophical dilemmas were thus raised:

1. Should the fixture be continuously adjusted during test to compensate for progressive compaction of the cushioning?

2. If not, would not the reduced transmission of high frequency vibration engendered by the loss of contact of the mass with the upper cushion make a fatigue-prone cushion appear more favorable than a less susceptible material?

3. If the degeneration of the cushion is progressive during the test cycle, are the responses observed during the latter phases pertinent to the material in its virgin state?

These questions must be considered in relation to the original test objectives. The transmissibility test was not predicated as a test of the vibrational integrity of the materials tested, but as an evaluation of their effectiveness as isolators (and minimal amplifiers) of vibration. Thus the cushioned item, although it appears in a very simplified form, is the true focus of concern in this test. Any change in the character of the cushioning during the test must be regarded as a serious dilution of the original test objectives.

Point-by-point excitation of frequencies closely approximates the original concept of continuous, single-frequency disturbances. However, this procedure tends to be quite time-consuming, and the resulting high fatigue potential is an obstacle to reproducible, unequivocal response data. The implied alternative to point-by-point programming of frequency is the continuous variation of frequency with simultaneous recording of response, i. e., "sweep frequency testing." However, the entire concept of the response function and transmissibility was predicated on the "steady-state" response to continuous functions. Before accepting "frequency sweeps" as a superior procedure, the relationship of the responses thus stimulated at specific frequencies to those resulting from "fixed frequency programming" must be considered.

Subjectively, it would seem reasonable that the response to a slowly changing frequency with constant amplitude would approach that of a fixed amplitude and frequency excitation. However, from the standpoint of minimizing cushion fatigue, a rapid rate of frequency change (sweep rate) is desirable. These conflicting objectives must be reconciled. Specifically, it is desired to establish the maximum frequency sweep rate that will produce a response of the cushion system equal to that produced by fixed frequency excitation. This area was clarified by the studies of F. M. Lewis, which are summarized in References 5 and 6. The maximum instantaneous sweep rate, df/dt, that would allow all resonances in a given frequence range f_1 to f_2 to reach their essential steady state amplifications was shown to be: $df/dt = f^2/R$. R is a constant dependent on the amplification of the resonances. This may be readily apprehended. The resonant peak for a high amplification system is relatively narrow. Thus, if it is approached in frequency too rapidly, full amplification does not take place. The allowable sweep rate is seen to be inversely proportional to system amplification. The effect of sweep rate is pictured in Figure 20 (p 50). The heavy curve is the theoretical response curve. The lighter curves are "apparent" transmissibility curves obtained with specific values of the parameter, R. The dotted curves illustrate that the response curve is the envelope of peak sinusoidal response as the frequency varies. Previous tests of complete cushioned packages indicate that resonant transmissibilities in excess of 15 are unlikely to be encountered since the probability of eventual nonlinear effects increases as resonant amplification increases. (Figures 18 and 19 reveal the amplification limiting action of nonlinearity.) From these considerations, and examination of Figure 20, the use of a value of R = 400 to calculate rates of sweep for cushion testing would seem reasonable. Integration of the relationship for instantaneous sweep rate yields the incremental time for traversing a frequency band f_1 to f_2:

$$\Delta t = [(f_2 - f_1)/f_2 f_1] \times R.$$

The minimum times for sweeping each of the acceleration vs frequency excitation spectra of Figure 10 may thus be defined:

Acceleration, g	Frequency Band, (cps)	Sweep Time, (seconds)
5	50 - 300	6.7
4	45 - 300	7.9
3	40 - 300	8.7
2	30 - 300	12.0
1.3	5 - 300	78.8
1	4.5 - 300	87.5
½	3 - 300	132.0
¼	2 - 300	196.0

The validity of these minimum time intervals depends on the achievement of a frequency sweep rate changing rapidly with frequency within the interval. Therefore, to fully realize the possible reduction in transmissibility test time without impairing the accuracy of the response data, automatic programming of excitation frequency is desirable.

Sweep frequency testing involves continuous recording of response data. While oscillographic recording may be used, the reduction in test time relative to manual programming and data recording is accompanied by increased data reduction time. The direct recording of response vs frequency on an X-Y recorder (utilizing appropriate converters) minimizes this problem.

Automatic frequency programming and response recording are being incorporated into the Picatinny facility and will be used in subsequent transmissibility testing.

The use of sweep frequency testing for the experimental definition of the response curves of nonlinear cushion systems may involve a complicating effect known as the "jump" phenomenon. While its influence on test results is readily rationalized, failure to consider this phenomenon may introduce error into the determination of resonant frequencies and maximum amplifications. A characteristic response curve for a softening system is portrayed in Figure 21 (p 51). Assume that the exciting frequency is being swept upwards while a constant excitation amplitude is maintained and the response is continuously recorded on an oscillograph or X-Y recorder. The recorded responses would increase along the "lower" path of the steady-state response curve until the slope (on the X-Y plot) reached 90°. A further increase in frequency would cause the response to increase abruptly (hence "jump") to the value on the "upper" path. Upward sweeping of the frequency beyond the jump frequency would reproduce the remainder of the steady-state response curve. From inspection of the response curve taken in the upward direction, one might erroneously conclude that the jump frequency was the resonant frequency, i.e., the frequency of maximum response. However, a higher level of response could be established at a lower frequency under steady-state excitation. This response condition can also be established in a downward frequency sweep from above the resonant frequency, as Figure 21 indicates. As the frequency is swept below the resonant peak, a response jump to the lower path must occur. By sweeping frequency in both directions, all possible vibratory response states are excited except those represented by the dotted curve in Figure 21. These vibrations are unstable and unlikely to persist in the environment.

The progressive deterioration of materials under cyclical mechanical loading is generally categorized as fatigue. For a tension-loaded material, the culmination of the deterioration in failure of the material may logically be defined to occur on total rupture and separation of the load-bearing element. Comparison of the tensile vibration resistance of similarly loaded materials may thus be conveniently based on the number of cycles of a specific stress required to induce failure. For a specific cycling rate, the criterion becomes the time to failure.

In the case of compression-loaded cushioning, no simple criteria for failure are available. Determination of the fatigue resistances of such materials, using time to failure as an index, would necessitate the establishment of an arbitrary description of cushion failure. The parameter chosen would necessarily be a simple one (such as cushion thickness) that could be conveniently monitored during the conduct of the vibration fatigue test. However, the relation of a simple visual index of failure, such as thickness loss, to the degradation of specific cushioning properties may be expected to vary widely for different types of cushioning. (That actual fatigue susceptibility of cushioning materials may vary widely was demonstrated in the preliminary transmissibility tests.) Thus the time-to-failure criterion for the evaluation of cushion vibration resistance is difficult to implement rationally and involves controversial standards of quality.

The alternative to a test of indeterminate length based on arbitrary standards of failure is seen as a vibration test of predetermined length and severity, followed by evaluation of changes induced by the test in previously defined cushion properties. The most important properties to be considered are 1) Thickness, 2) Modulus, 3) Shock Transmission, and 4) Resonant Amplification. Thus the pertinent parameters for evaluating the relative adequacy of various cushions for a specific application will be available to the packaging engineer. The arbitrary loading cycle to be utilized as the "fatigue input" will be the ASTM vibration envelope swept repetitively for 1 hour at a specific logarithmic frequency sweep rate (automatically programmed). Vibration response will be recorded at specific intervals to define item 4 above. Items 1 and 2 will be measured before and after the fatigue test. Impact testing will be performed per ASTM D-1596-59T on unvibrated cushions. Fatigue tests will be conducted on virgin specimens at static stresses in the "optimum range" established by the above procedure. After the fatigue test, the vibrated cushions will be subjected to the ASTM impact test at the same bearing stresses used in the fatigue test. Degradation of shock-mitigating properties and impact set resistance produced by the fatigue cycle are regarded as significant criteria for defining the fatigue susceptibility of cushioning media.

TEST PROCEDURE FOR PRELIMINARY TESTING

General transmissibility test procedures were based on the "Proposed (ASTM) Standard Method of Test to Determine the Vibration Transmission Characteristics of Package Cushioning Materials" (Appendix A). The test configuration conformed to paragraph 5A, "Compression Configuration" (see Fig 1, p 33).

1. The resilient polystyrene foam was cut into 14 test specimens, each test specimen measuring approximately 10" × 10" × 5" (100 square inches bearing area). The resilient polyurethane foam was cut into 2 samples, each 10" × 10" × 3" (100 square inches bearing area).

2. All specimens were measured, weighed, and their density calculated.

3. Compression tests were performed on all samples to 15% strain and the stress-strain relationship recorded.

4. Seven pairs of polystyrene specimens were matched for approximate density.

5. The cushion pairs were assembled into the "guide fixture" as shown in Figure 1. Double-backed tape was used to attach the upper and lower cushions to the top plate and guide fixture, respectively. Auxiliary weights were bolted to the guide weight to produce the static bearing stress on each bottom cushion indicated in Table 1 (p 32). The top plate was placed over the top cushion so as to impose its weight (22 lb) on the upper and

lower cushions, and was then rigidly bolted to the guide fixture by means of 4 bolts, so as to constitute a single-degree-of-freedom package.

6. This package, rigidly fastened to the vibration machine, was excited with sinusoidal vibration at the peak acceleration levels and frequency ranges indicated in Table 1. The vibrator was set at specific frequencies in each frequency range and the excitation brought up to the specified acceleration. The output of the response accelerometer was monitored on the calibrated oscilloscope (Fig 5, p 37) and the peak acceleration response noted at each set point. (Approximately 25 frequency set points were made at each excitation level. Each frequency scan at a specific acceleration level required between 10 and 15 minutes.) The peak response for each level of acceleration is recorded in Table 1. Overall response curves appear in Figures 26 through 29 (pp 56 through 59).

7. Subsequent to the transmissibility tests, the thickness of each specimen was again measured.

8. Compression tests were repeated to 15% strain on the vibrated specimens.

9. The vibrated polystyrene cushions were then impact-tested in accordance with ASTM D 1596-59T. Five impacts were imparted to each cushion to obtain transmitted-acceleration and impact-set data corresponding to that reported for unvibrated polystyrene cushions in Picatinny Arsenal Technical Report 3017, "Dynamic Cushioning Properties of Resilient Polystyrene Foam." Impacts were made at velocities corresponding to 3-foot free falls under specific static stresses corresponding to those used in the transmissibility testing.

10. The averages of the transmitted peak accelerations and the cumulative impact set (%) for 5 impacts are given for vibrated and unvibrated cushions in Table 1. The data for the unvibrated cushions is derived from Figures 23 through 25 (pp 53 through 55), which are taken from the report referenced above.

DISCUSSION OF TEST RESULTS

The objectives of the preliminary vibration testing were the evaluation of the concepts, techniques, and equipment for vibration transmissibility testing that are described in preceding sections of this report. Specifically, it was desired to accumulate steady-state response data for comparison with similar data to be obtained with programmed sweep equipment being incorporated into the test facility. The relative validity of the two approaches to response determination had been a source of considerable debate in the ASTM task group formulating test cirteria in this area. Whereas the point-by-point method as used in the preliminary testing was recognized as adhering closely to the fundamental concept of continuous forcing functions and steady-state response, concern over the reproducibility of test results centered on the indeterminate time scales associated with its implementation (with attendant variability in fatigue potential). The preliminary testing showed this concern to be well founded--to a totally unanticipated degree.

Resilient polystyrene foam was selected for these tests since its shock-transmitting qualities as a function of static stress had been investigated and reported in a previous Arsenal report. Vibration testing could thus be concentrated on the loading range within which its efficiency as a shock isolator had been shown to be maximal. The cushioning efficiency of this material is maximal over a relatively wide range of static bearing stresses for a cushioning material (0.6 to 1.5 psi) and is relatively constant over the military environmental temperature range (-65°F to +160°F). Upon determination of pertinent vibration response data, a comprehensive data package could be compiled for this material.

The initial intent was to excite each cushion pair with all the scheduled vibration levels of Figure 10 (0.25 to 5 g's) under a specific bearing stress. For each density of the material, several bearing stresses blanketing the optimum load range for that density were to be employed, new cushion pairs being used for each bearing stress condition. To minimize the effects of fatigue on the transmissibility data, the response tests for each cushion pair were to be conducted in the order of increasing intensity of excitation.

Maximum usable bearing stress for the polystyrene was assumed to be enough stress to produce 10% cumulative set in 5 equivalent 36-inch free-fall impacts. For the initial density tested (0.8 lb/cu ft), reference to Figure 24 (p 54) indicates this stress to be 1.5 psi. Table 1 (p 32) indicates that thickness losses of 18% and 36% in the upper and lower cushions, respectively, were induced at this bearing stress by the lowest excitation levels, 0.25 g's and 0.5 g's. The thickness losses are shown pictorially in Figure 6 (p 38). (The accumulation of data points occupied approximately 15 minutes, about 5 minutes of which was spent near the resonant frequency, 8 cps.) The degree of compaction experienced was regarded as precluding both numerically valid response data and reasonable application value. Subsequent tests were therefore conducted at bearing stresses near the lower end of the optimum load range for the 3 densities tested. Even at the reduced bearing stress, the vibration set, although reduced, generally exceeded the 10% limit for impact set in the 5-impact sequence. The 10% set condition was found to be reached after excitation at even the lowest levels, and the excitation of the complete schedule of vibration on each cushion pair was therefore waived.

The vibration response curves generated by the testing of the polysytrene foams appear in Figures 26 through 29 (pp 56 through 59). In view of the general progressive degeneration experienced during acquisition of the data, the curves are presented for discussion purposes only. While the entire response curve for a particular condition of bearing stress and excitation level may not be pertinent to the material in a virgin state, general conclusions as to the vibratory response of the material may be derived from the trends of the data. Additional insights may be obtained from the partial force-deflection curves for each test sample taken before and after vibration testing. The post-vibration force-deflection curve for each cushion is shown offset by the amount of the vibratory set in order to convey more fully the implication of the changes in energy-absorbing capabilities. The before-and-after compression characteristics of the fourteen samples tested are shown in Figures 31 and 32 (pp 61 and 62).

Reference to the force-deflection curves for samples 1 through 4 shows the initial stiffness (the ratio of incremental force to incremental deflection; i.e., the slope of the force vs deflection curve) for the 0.8 lb/cu ft polystyrene foam to be approximately linear before vibration. Vibration of cushions 1 and 2 with 0.25 and 0.5 g's excitation under 1.5 psi bearing stress (No. 1, bottom cushion) produced thickness losses of 1.8 and 0.9 inches in cushions 1 and 2, respectively. Figure 31 indicates that, despite the loss of thickness during vibration, the cushion stiffness did not change materially. The resonant frequency for a cushion of linear stiffness was shown previously to be independent of excitation amplitude. Figure 26 verifies that essentially linear dynamic stiffness was operative in the 0.8 lb/cu ft foam under 1.5 psi bearing stress since the indicated resonant frequency is 8 cps for both 0.25 and 0.5 g excitation. The amplification at resonance for 0.5 g excitation is seen to be 6.6 as compared to 8.5 for 0.25 g input, indicating that the damping mechanism is nonlinear and hence predominantly hysteretic rather than viscous in nature.

The reduction of the static stress for cushions 3 and 4 to 0.7 psi is observed (Fig 27, p 57) to have increased resonant frequency from 8 cps to 11 cps. This shift in resonant frequency agrees with linear theory prediction. The amplifications at resonance at the reduced stress (8.0 for 0.25 g excitation and 7.7 for 0.5 g excitation) do not display the increase in resonant response that would take place in a viscously damped cushion; again, this indicates the damping is nonlinear.

The reduction in static stress on the 0.8 lb/cu ft cushion from 1.5 to 0.7 psi is seen to markedly reduce the vibration set in both the upper (2 and 3) and lower (1 and 4) cushions (Table 1 and Fig 31). While the maximum accelerations transmitted were not significantly altered by reducing the static stress, the maximum dynamic stresses imposed on both the upper and lower cushions were considerably reduced, the maximum stress in the bottom cushion being $(g + 1) \times$ static stress and that in the top cushion $g \times$ static stress. Thus for 0.5 g excitation, reducing the bearing stress from 1.5 to 0.7 psi would reduce the peak stress in the bottom cushion from 6.3 psi to 3.4 psi, and in the top cushion from 4.9 to 2.7 psi. Table 1 indicates that the thickness loss in the bottom cushion is reduced from 36% to 10% and in the top cushion from 18% to 8%. While the time scales for the conduct of the response tests were subject to only approximate control, the disparity in vibratory set is of sufficient magnitude to emphasize the strongly determinant aspect of bearing stress in the fatigue susceptibility of polystyrene foam.

In the response tests of the 0.5 lb/cu ft polystyrene, the bearing stress was lowered to 0.6 psi since the impact tests (Fig 23) indicated a somewhat lower optimum loading range, and a susceptibility to rupture under higher bearing stresses. To minimize the effect of fatigue on vibration response, only a single level of excitation was imposed on each cushion. The force-deflection data for cushions 5 through 10 (Fig 31 and 32) indicates that this characteristic relationship for the 0.5 lb/cu ft material was substantially altered by the response testing despite this precaution. The force-deflection curves for the unvibrated cushions manifest a distinctly bilinear characteristic; a distinct "softening" is seen to occur at approximately 0.1-inch deflection. The response curves at 0.25 and 0.5 g's excitation (Fig 28) display the characteristic downward shift of resonance with forcing amplitude typical of a "softening" system. The data points for these two levels of excitation are seen from Table 1 to have been obtained in decreasing frequency increments--the valid procedure for obtaining the maximum response of softening systems. The response curve for the

1.0-g excitation level does not follow the trend established by the two lower levels of forcing vibration, in terms of resonant frequency shift. The reasons for the apparent anomaly may be discerned from Figures 31 and 32. The "break-point" in the force-deflection curves for cushions 5 and 6, which were subjected to the 1.0 g vibration, is observed to occur at a higher force level than the break-points for the other cushions in the 0.5 lb/cu ft group (Nos. 7 through 10). The maximum dynamic force, as determined from the response data, indicates that less force was developed than would be needed to reach the break-point in the force-deflection curve for the unvibrated cushion. If the force vs deflection curve taken on the unvibrated cushion was the controlling function through resonance, the indicated resonant frequency would be expected to be closer to that established under the 0.25-g vibratory input. However, reference to the post-vibration force vs deflection curve indicates that the bilinear characteristic of the new cushions is drastically altered by vibratory fatigue. The initial slope in the fatigued cushions is seen to correspond approximately to the slope in unvibrated cushions beyond the break point. The major part of the thickness loss was observed to take place during vibration at and near resonance, as might reasonably be expected. The vibration response data for the 1.0 g excitation of the 0.5 lb/cu ft polystyrene was taken in the direction of increasing frequency, starting below the resonant frequency and proceeding through resonance to the higher frequencies. It is reasonable to believe that the controlling force-deflection relationship at resonance would be intermediate between that obtained for new cushions and that obtained at the end of the 1.0 g response study. The reduction in transmissibility of higher frequency vibration that occurred at 1.0 g input (Fig 28) is also attributed to the cushion fatigue involved in obtaining response data through resonance. Cushions 5 and 6 were "softened" by resonant vibration before they were subjected to the higher frequency vibrations, whereas the other cushions of the group were exposed to the high frequency excitation in the stiffer virgin state. Thus a lower resonant frequency system (with its attendantly greater attenuation of high frequency vibration) was operative at high frequency at 1.0 g excitation than at the other excitation levels.

The response curves shown in Figure 29 for the 1.1-lb/cu ft polystyrene are indicative of a "softening" system. The decreasing slope of the force-deflection curves for this density is apparent in most of the cushions in Figure 32. The softening characteristic is seen to be accentuated after the conduct of the vibration surveys. When loaded at 0.6 psi, the 1.1-lb/cu ft polystyrene evidenced the least degradation of physical parameters as a result of vibration of the three densities tested. It is noted, however, that under 1.0 g excitation the thickness loss exceeded 10%, a figure that had been fixed as a limiting value for the usefulness of materials as shock-mitigating media. Figure 10A (p 41) indicates that a 1.0 g level of vibration does not represent an excessively severe simulation of "rough-road" vehicular vibration. The approximate duration of the response studies (15 minutes) would not appear to be an overly long excitation interval in terms of normal logistical patterns.

To determine whether point-by-point evaluation of response is feasible for any commonly applied cushioning media, response tests were made on a 2 lb/cu ft, 3-inch-thick polyether urethane foam. The maximum bearing stress recommended for this material was 0.16 psi (fixed by creep considerations). Response tests were conducted under this bearing stress at 0.25, 0.5, 1.0, 2.0, and 3.0 g's dynamic excitation. The same pair of cushions was used for all excitation levels. No measurable thickness loss resulted from this procedure. The response curves obtained are plotted in Figure 30. With the initial increase in excitation level, the resonant frequency is observed to shift to a lower frequency. Increase of excitation from 1.0 g to 2.0 g's is observed to induce a shift upward in resonant frequency. These effects were predicted earlier in the report on the basis of the stress-strain curve for this material (Fig 17). Calculation of maximum dynamic stresses from the response curves indicates that the resonant responses for 0.5 and 1.0 g excitation lie in the "softening" portion of the stress-strain curve, while the response to the 2.0-g input lies in the "stiffening" region at approximately 70% strain.

Maximum response at 1.0 g excitation is seen to be 1.5 g's for the polyurethane, and 4.5 g's and 6.0 g's for 0.5- and 1.1-lb/cu ft polystyrenes, respectively. Maximum g's for the polyurethane with 0.25 g input is 0.75 g while the average resonant response of the various polystyrene configurations to this input is 1.6 g's.

In comparing these two types of cushioning media for overall performance in a vibration environment within the stress levels tested, the superiority of the polyurethane is evident, both in low resonant transmissibility and low fatigue susceptibility. While the resonant frequencies of the two materials in the configurations tested were not markedly different from each other, direct comparison of high frequency transmissibility is not regarded as meaningful (because of cushion fatigue) since the data on the polystyrenes was taken in the increasing frequency direction. Data taken in the downward frequency direction would tend to indicate a somewhat higher transmissibility in the polyurethane at high frequencies because of its high indicated damping. (Apparently, the damping is viscous in nature.)

In terms of excitation levels and time involved, the response studies of the resilient polystyrene did not approach the severity of the standard fatigue test proposed earlier (cycling of the "outer" ASTM envelope for 1 hour). However, the magnitude of the compaction experienced by the foam, and the strongly determinant aspect of bearing stress in this fatigue phenomenon, led to the decision to evaluate the effect of the limited vibration response studies on the shock transmitting properties of the polystyrene foam. A comparison of results for vibrated specimens with results for similar unvibrated cushions (Ref 7) appears in Table 1. The impact recovery properties of the polystyrene foam appear to have been almost totally destroyed by the fatiguing effect of the vibration response tests at 1.5 psi bearing stress. The impact set for the cushions vibrated under 0.7 psi stress is seen to be markedly less (58% as against 78% for the 1.5 psi stress conditions). The 78% set value was induced by a 3-impact sequence as against 5 impacts for the 58% value. (In Table 1, the higher values of set obtained in both vibration tests and post-vibration impact tests apply to the lower cushion of each vibration pair.) The loss of impact set resistance caused by the vibration testing is dramatically visible in the photograph shown in Figure 8. As has been noted, the set in unvibrated cushions did not exceed 10% after 5 impacts at equivalent static loadings.

Table 1 indicates that the peak accelerations transmitted by unvibrated cushions of the same approximate density as test samples 1 through 4 averaged 38 g's at both 1.5 and 0.7 psi static stresses. Figure 24 shows these stress values to lie at the upper and lower ends of the optimum load range. While the vibratory set induced in the 1.5 psi stressed cushion would be expected to result in an increase in subsequent impact transmission, this result would not be automatically predicted for the 0.7 psi loading. The drastic increase in average acceleration transmitted in the 5-drop sequence resulted largely from the inability of the vibrated cushions to recover from the initial impacts. These observations tend to buttress a previously developed theory as to the nature of impact energy absorption in resilient polystyrene foams; i.e., impact energy is largely absorbed by compression of air trapped in the closed cell structure of the foam. Prior bases for this viewpoint were the observations that these foams are able to absorb large amounts of impact energy at high levels of dynamic stress. The same stresses, when applied at lower loading rates in a compression machine, produced greatly increased permanent set as compared to the dynamic application. This effect cannot reasonably be attributed to "dynamic stiffening" of the foam matrix, since the force-time histories and rebound considerations are indicative of an elastic rather than a viscous phenomenon. This theory implies that the resilience of these foams under impact is largely embodied in the contained air, rather than the cell structure. When the foam is vibrated under the high bearing stresses giving minimum shock transmission, the cell structure gradually deteriorates. The loss of cell integrity as a result of vibratory fatigue allows the contained air to escape on initial impacts after vibration. The inherent resilience of the matrix is insufficient to restore the cushion to its initial thickness. Successive impacts further compact the foam, with the result that the transmitted acceleration steadily increases.

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TABLE 1

Effects of vibration transmissibility testing on resilient polystyrene foam. Original thickness: 5 inches

Sample No.	Density, lb/cu ft	Static Stress, psi	Vib Input, g (pk)	Vib Resp, g (pk)[a]	Freq Range, cps	Impact g's[b] Unvib	Impact g's[b] Aft Vib	Thickness (inches) Vib	Thickness (inches) Vib/imp	Thickness (inches) Imp Only	Thickness Loss (%) Vib	Thickness Loss (%) Vib/Imp	Thickness Loss (%) Imp Only
1	0.8	1.46	0.25	2.13	2- 40	38	82.4[c]	3.2	1.1	4.7	36	78[c]	11
1	0.8	1.46	0.50	3.34	5- 50								
2	0.81	1.46	0.25	2.13	2- 40	38	57.2[c]	4.1	1.7	4.7	18	66[c]	11
2	0.81	1.46	0.50	3.34	5- 50								
3	0.84	0.71	0.25	2.0	2- 40	38	51.4	4.6	3.3	4.8	8	34	2.5
3	0.84	0.71	0.5	3.87	5- 50								
4	0.81	0.71	0.25	2.0	2- 40	36	67.0	4.5	2.1	4.8	10	58	2.5
4	0.81	0.71	0.50	3.87	5- 50								
5	0.46	0.60	1.0	4.5	10-100	40	48.4	4.2	3.1	4.6	16	38	7.5
6	0.46	0.60	1.0	4.5	10-100	40	49.0	4.6	3.7	4.6	8	26	7.5
7	0.46	0.60	0.25	1.38	40- 4	40	39.8	4.6	3.8	4.6	8	24	7.5
8	0.47	0.60	0.25	1.38	40- 4	40	41.4	4.7	3.9	4.6	6	22	7.5
9	0.48	0.60	0.50	2.8	50- 4	40	60.0	4.8	1.8[d]	4.6	4.0	64[d]	7.5
10	0.46	0.60	0.50	2.8	50- 4	40	42	4.7	3.7	4.6	6.0	26	7.5
23	1.1	0.60	0.25	0.95	2- 40	42	46	4.8	4.5	4.8	4.0	10	3.6
23	1.1	0.60	0.50	3.0	5- 50								
24	1.1	0.60	0.25	0.95	2- 40	42	48	4.9	4.6	4.8	2	8	3.6
24	1.1	0.60	0.50	3.0	5- 50								
25	1.1	0.60	1.0	6.0	100- 11	42	47.7	4.4	3.3	4.8	12	34	3.6
26	1.1	0.60	1.0	6.0	100- 11	42	36.0	4.7	4.7	4.8	6	6	3.6

[a] Peak mass acceleration measured in frequency range at indicated input level.

[b] Average of five 36" equivalent free-falls at indicated static stress. Data for unvibrated cushions from PA Tech Report 3017, *Dynamic Cushioning properties of Resilient Polystyrene Foam.*

[c] Data from 3 impacts. Compaction of cushion precluded 4th and 5th drops.

[d] Thickness data after 7 impacts. Acceleration indicated is average of first 5 impacts.

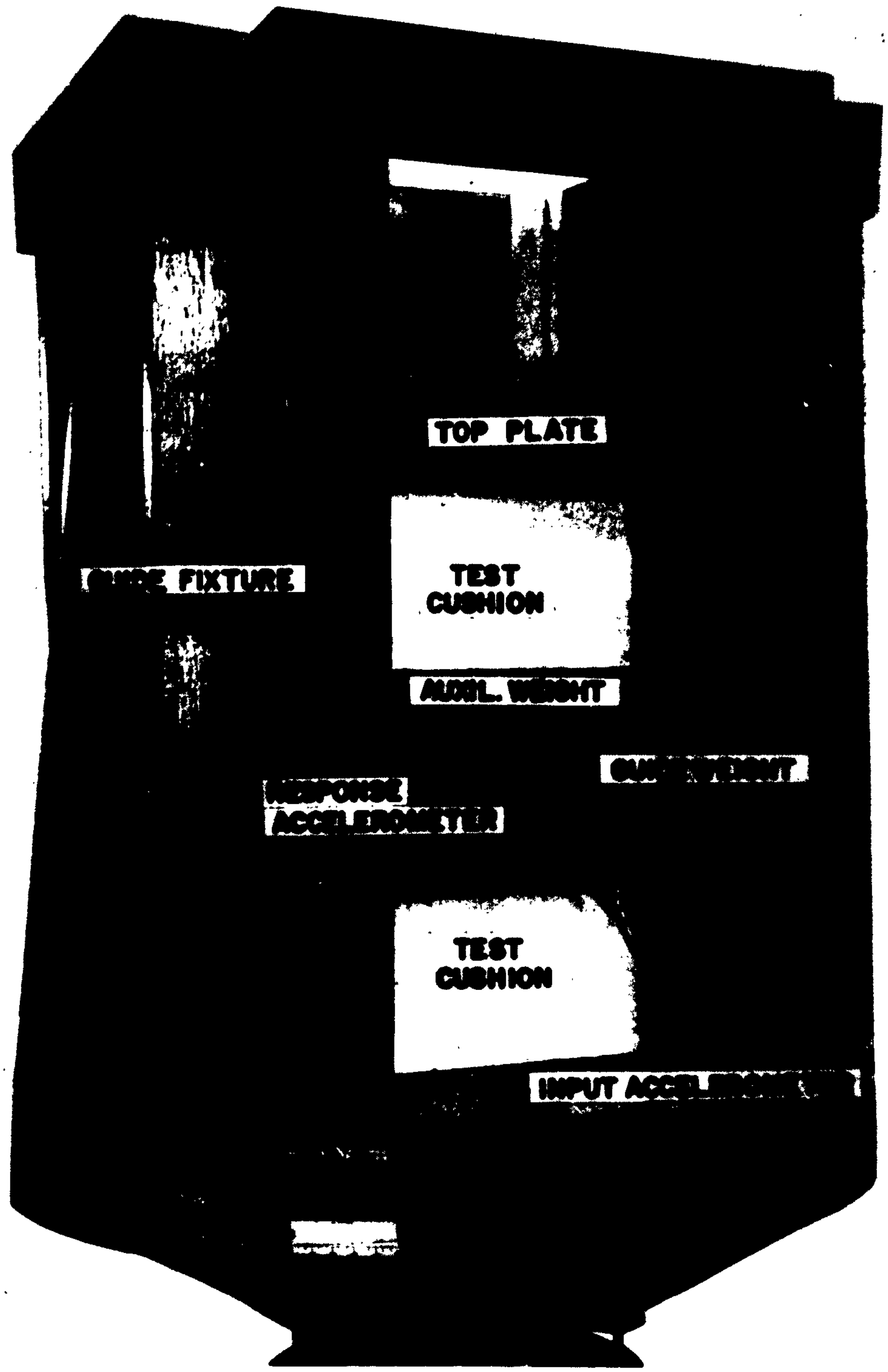

Fig 1 Vibration test assembly

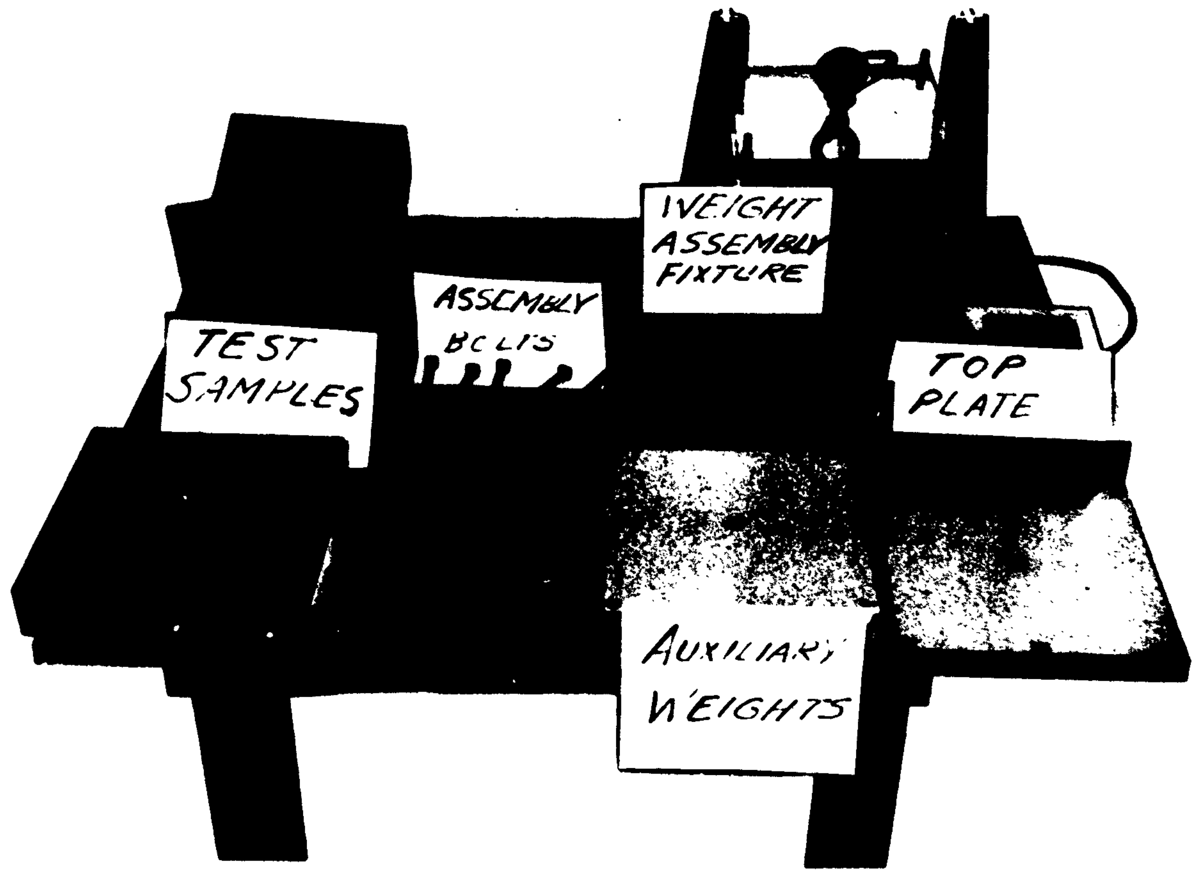

Fig 2 Elements of vibration test assembly

Fig 3   Guide fixture

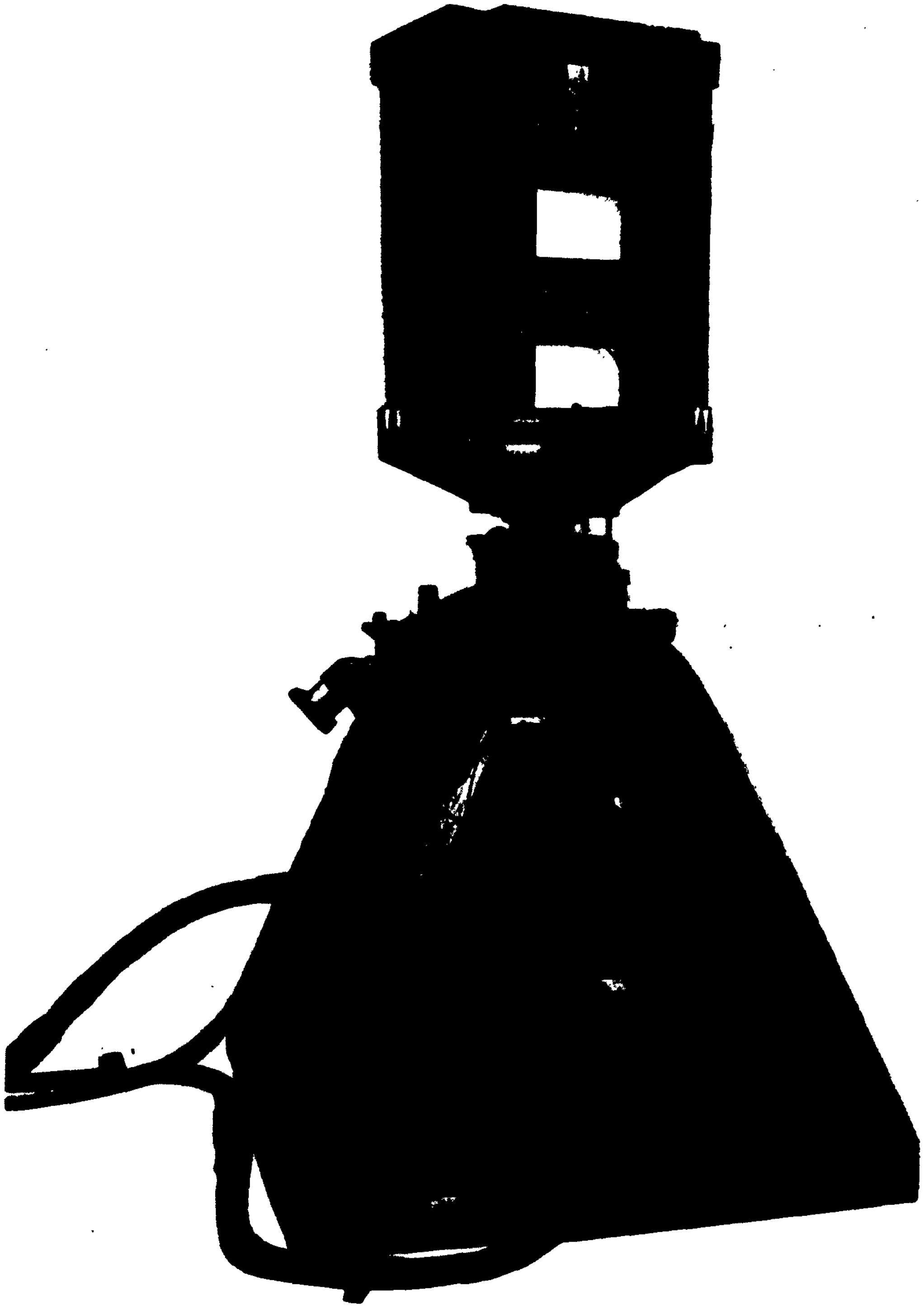

Fig 4 Vibration test assembly on hydraulic vibrator

1) OSCILLATOR (CALIBRATION)
2) CALIBRATION JUNCTION BOX
3) VACUUM TUBE VOLTMETER (CALIBRATION STANDARD)
4) ELECTROMETER AMPLIFIERS, D.C.
5) ELECTRONIC FILTER
6) OSCILLOSCOPE, D.C. 2 CHANNEL

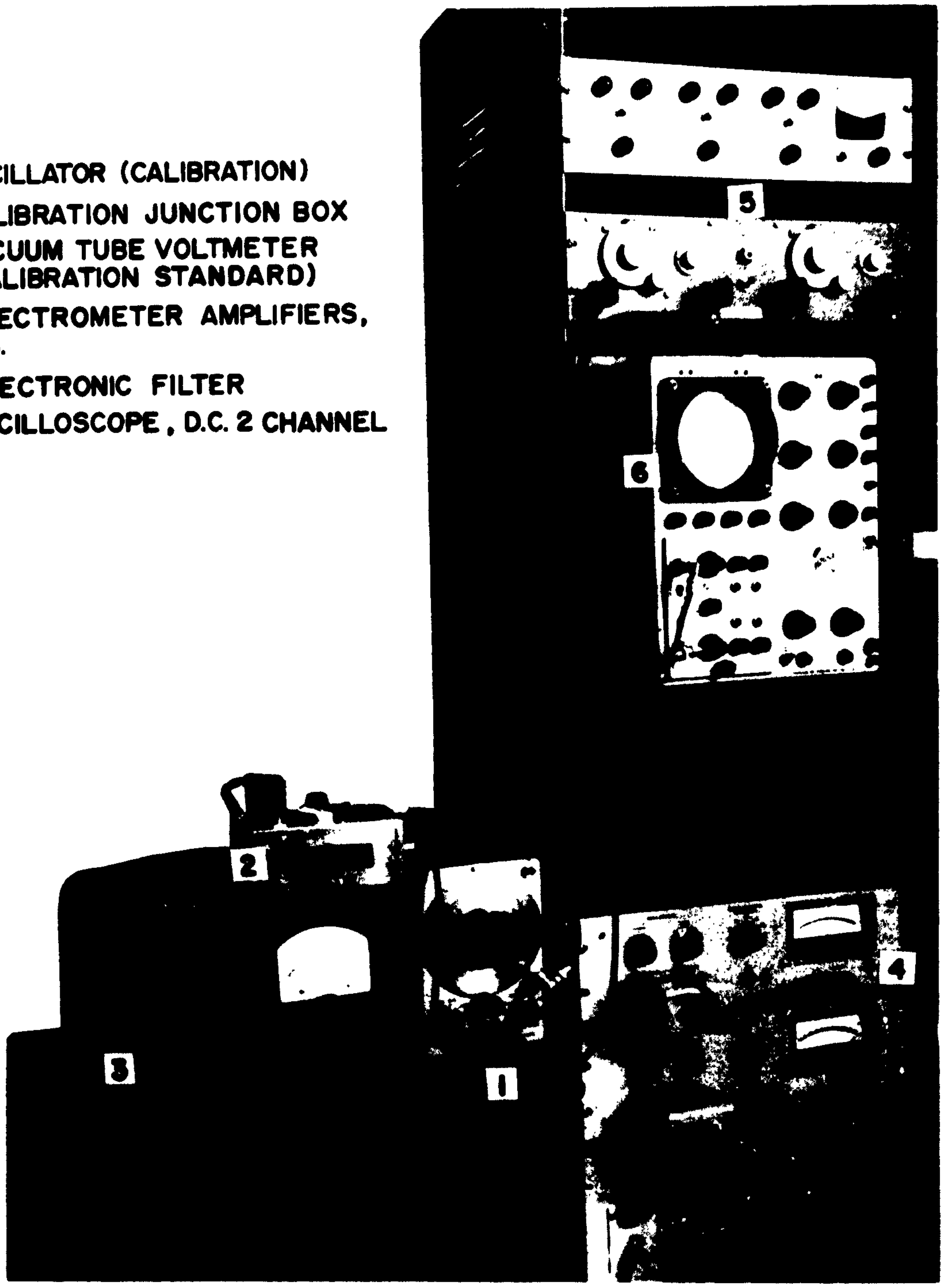

Fig 5 Instrumentation for acceleration measurement

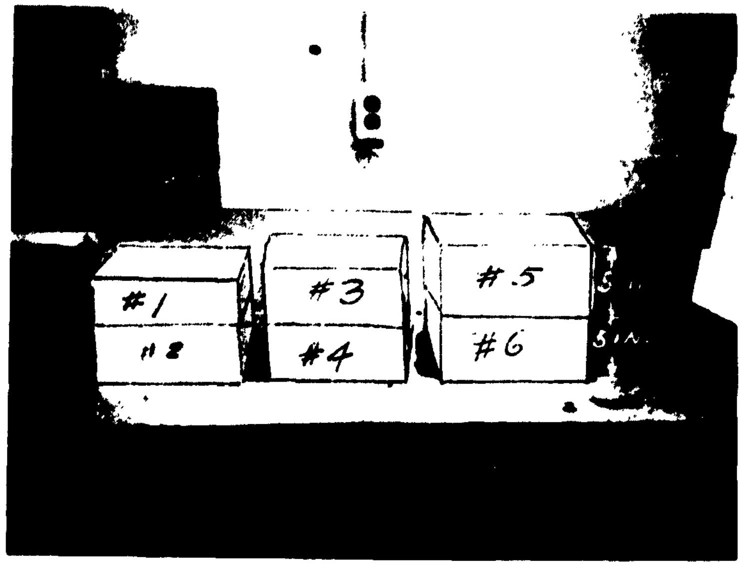

Fig 6

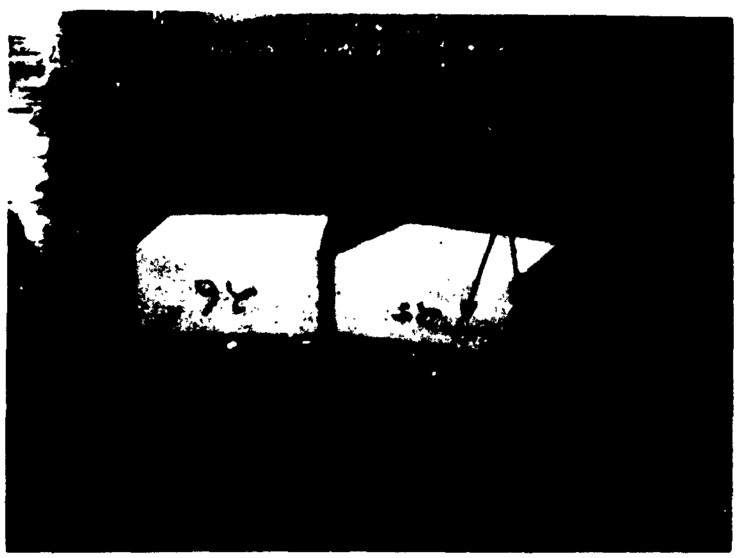

Fig 7

Permanent set of resilient polystyrene foam after vibration transmissibility testing

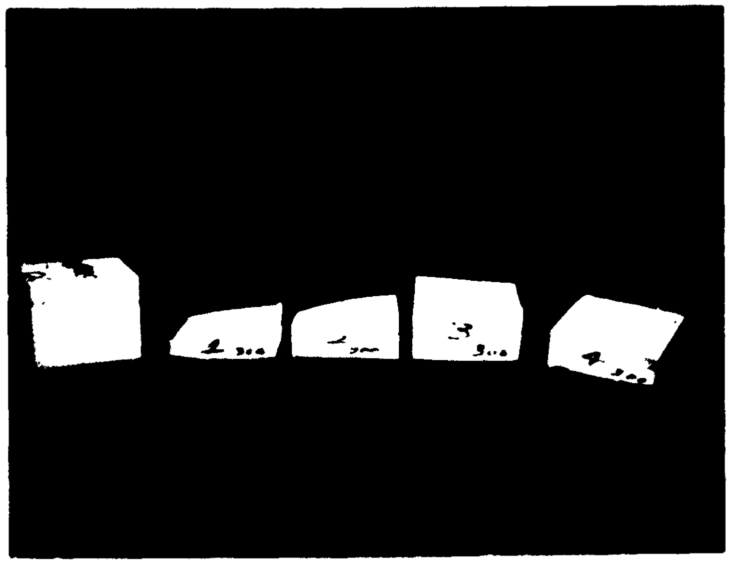

Fig 8

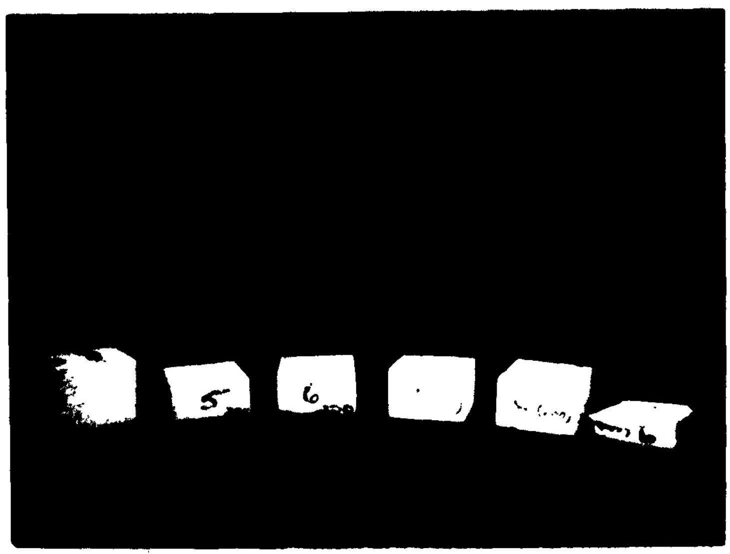

Fig 9

Cushions subjected to (5) impacts after transmissibility testing. (Original thickness at left. At same static loading, unvibrated cushions suffered less than 10% set after 5 impacts)

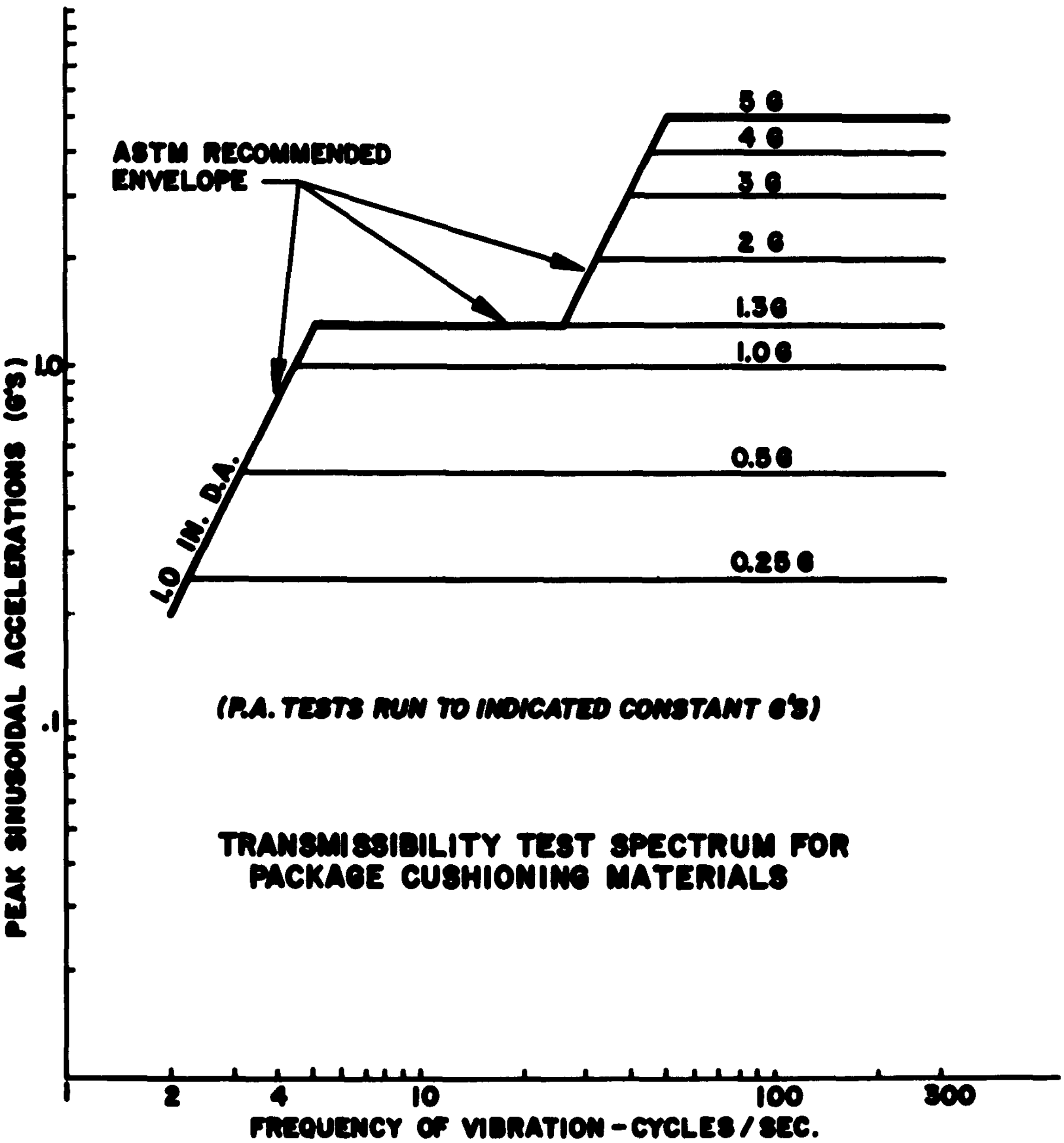

Fig 10 Transmissibility test spectrum for package cushion materials

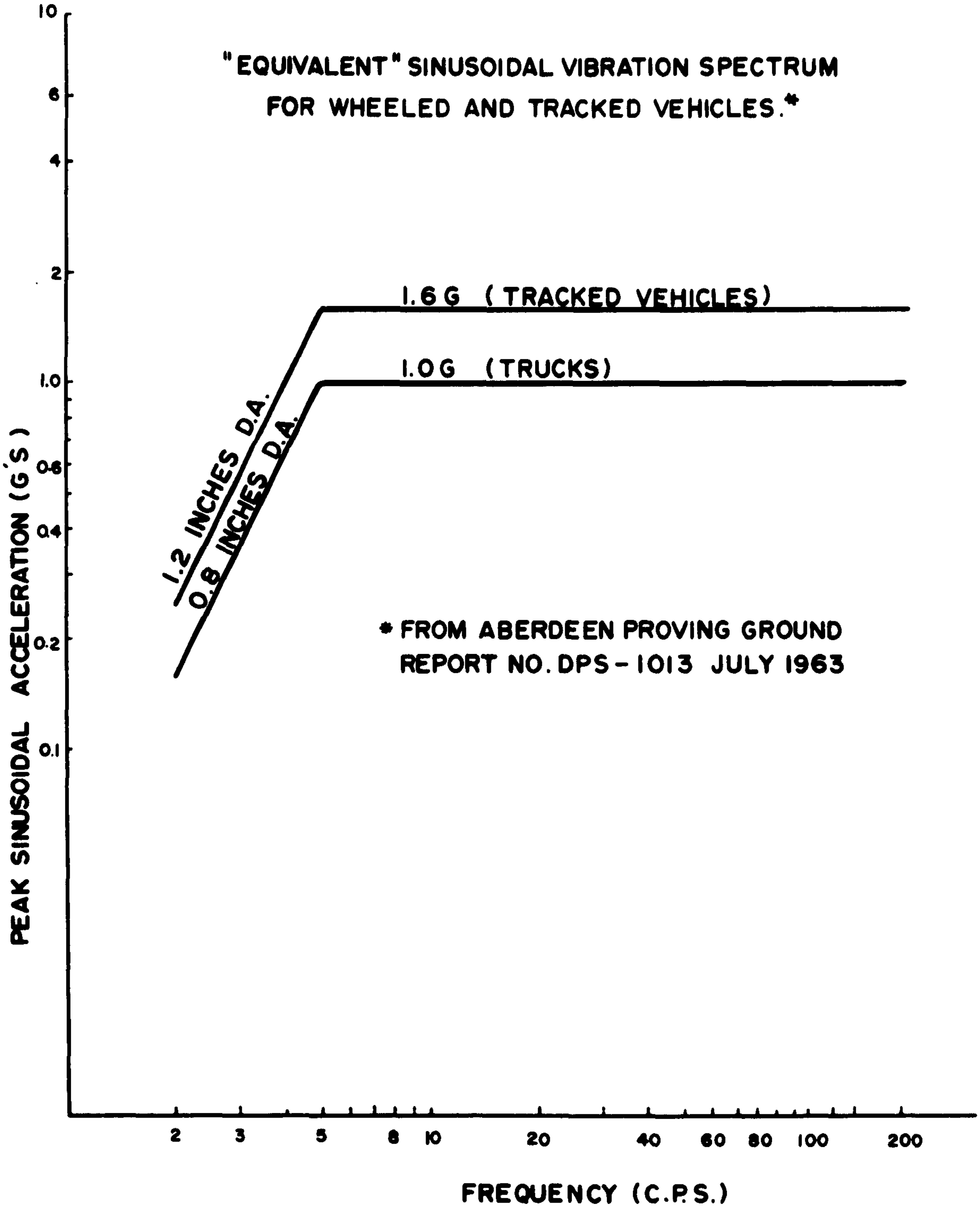

Fig 10A "Equivalent" sinusoidal vibration spectrum for wheeled and tracked vehicles

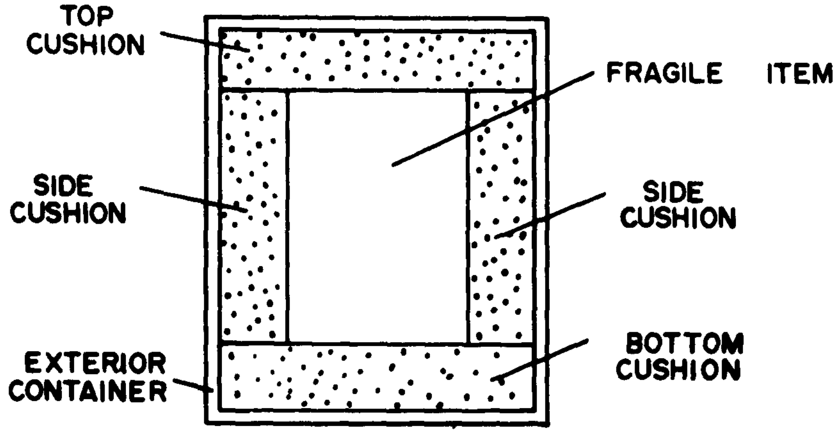

Fig 11 Schematic view of most common cushion application

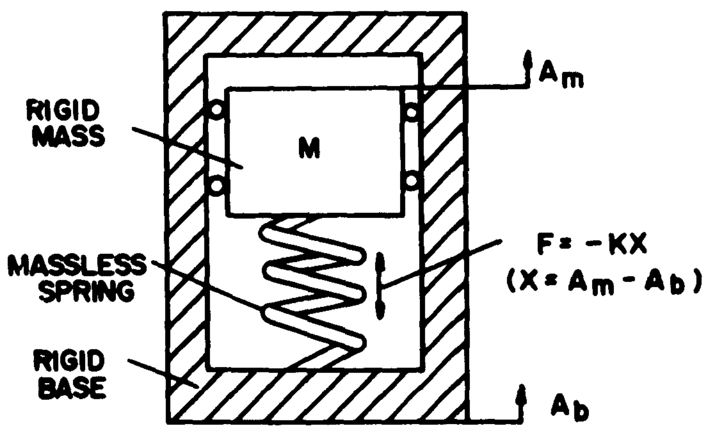

Fig 12 Classical single-degree-of-freedom system

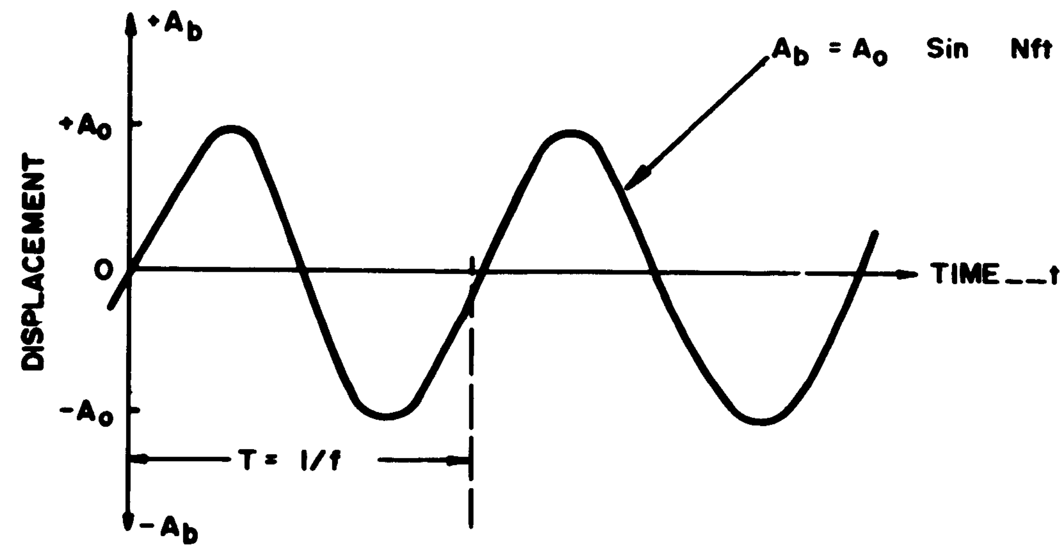

Fig 13 Sinusoidal displacement function plotted against time

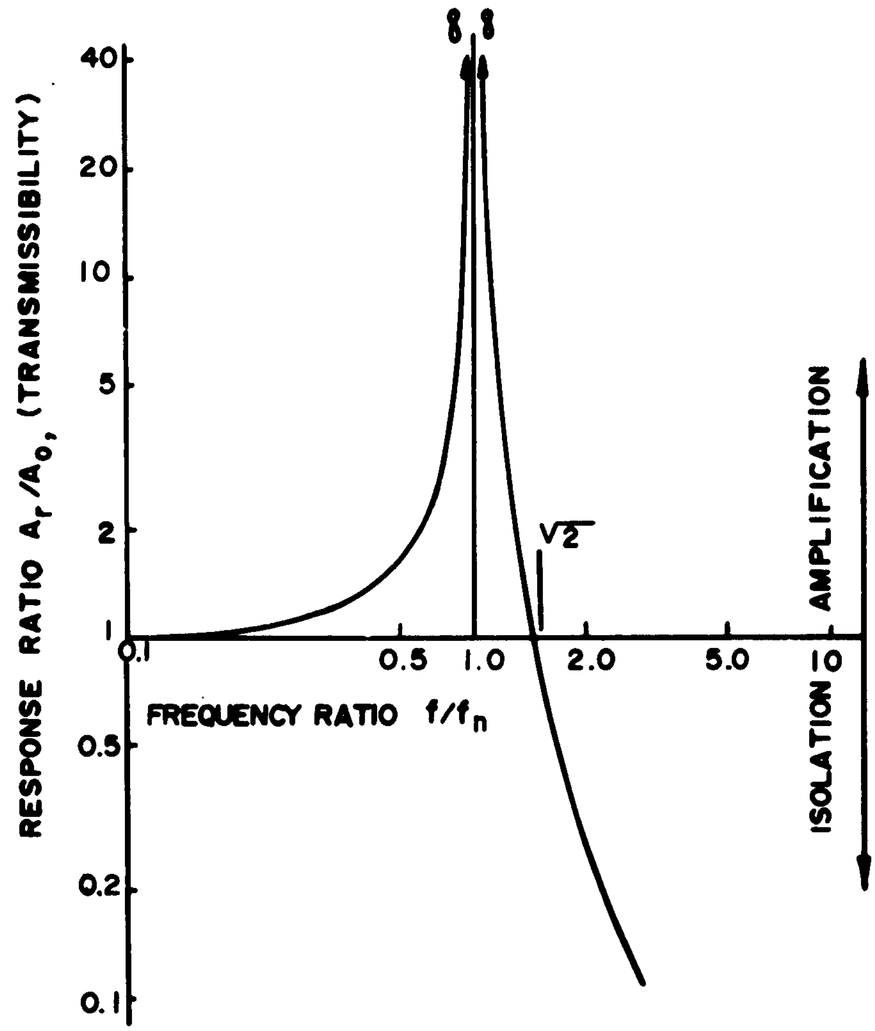

Fig 14 Vibration response ratio (transmissibility vs frequency) for undamped single-degree-of-freedom system

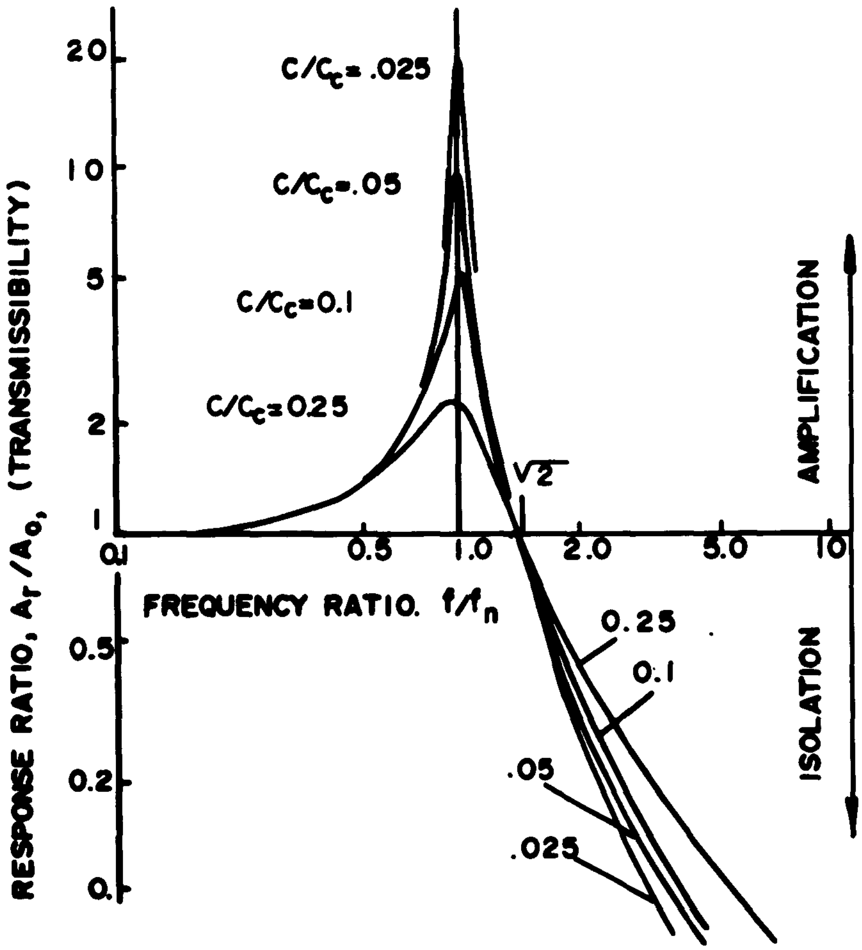

Fig 15 Vibration response ratio (transmissibility vs frequency) for viscous-damped single-degree-of-freedom system

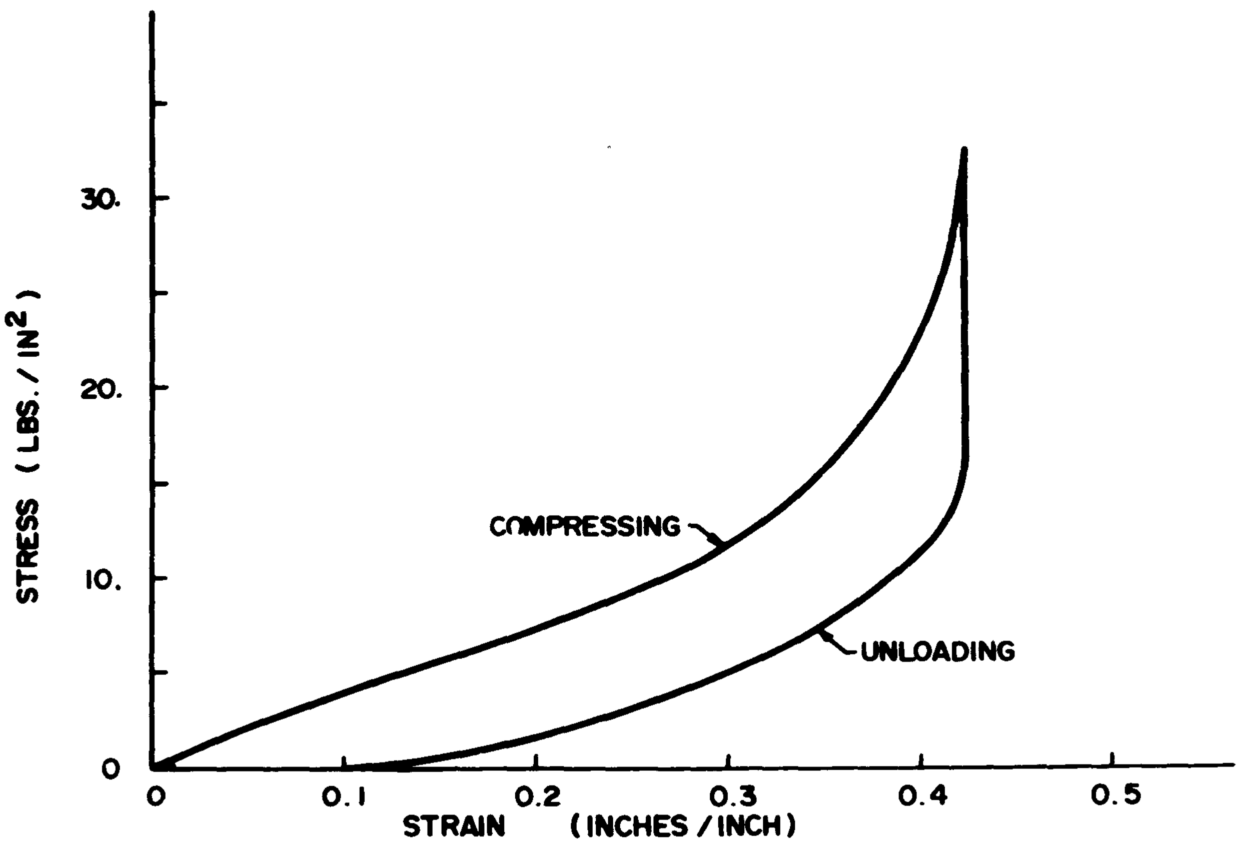

Fig 16 Stress-vs-strain curve for resilient polystyrene foam. Density: 0.5 lb/ft^3

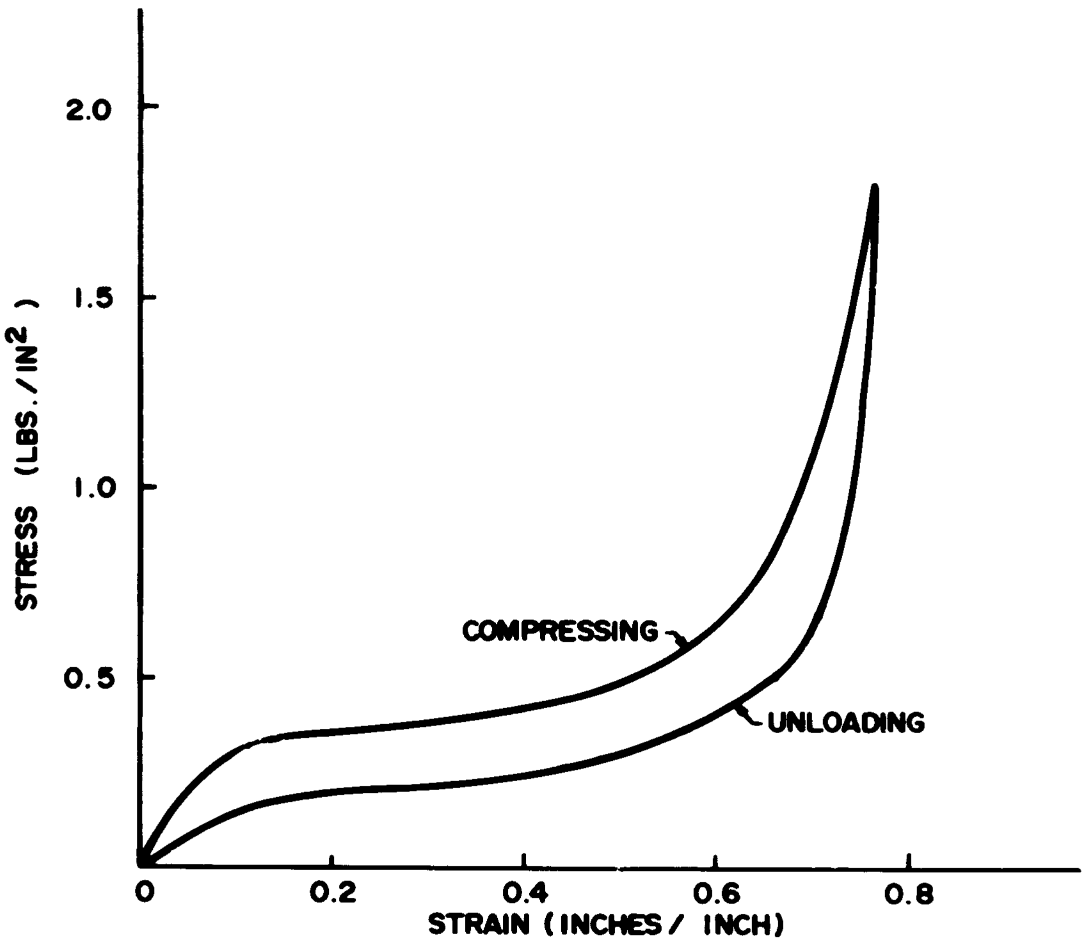

Fig 17 Stress-vs-strain curve for resilient polyether urethane foam. Density: 2.0 lb/ft^3

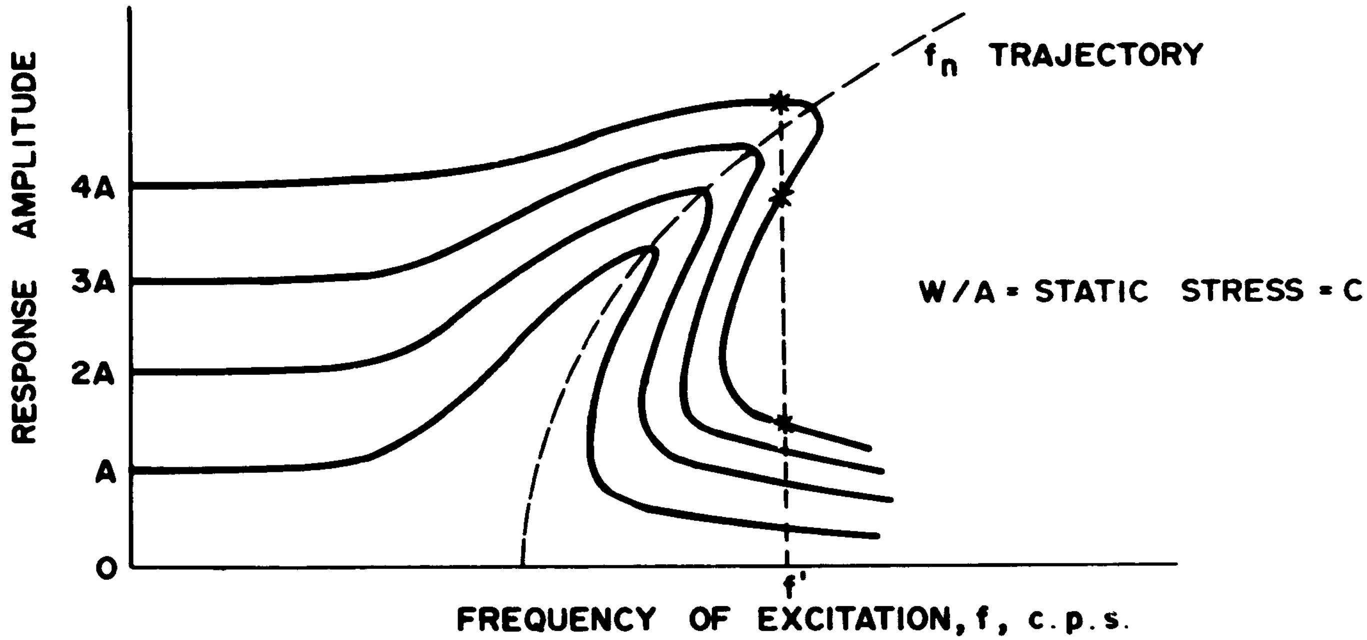

Fig 18 Response of nonlinear stiffening cushioning, showing effect of excitation amplitude

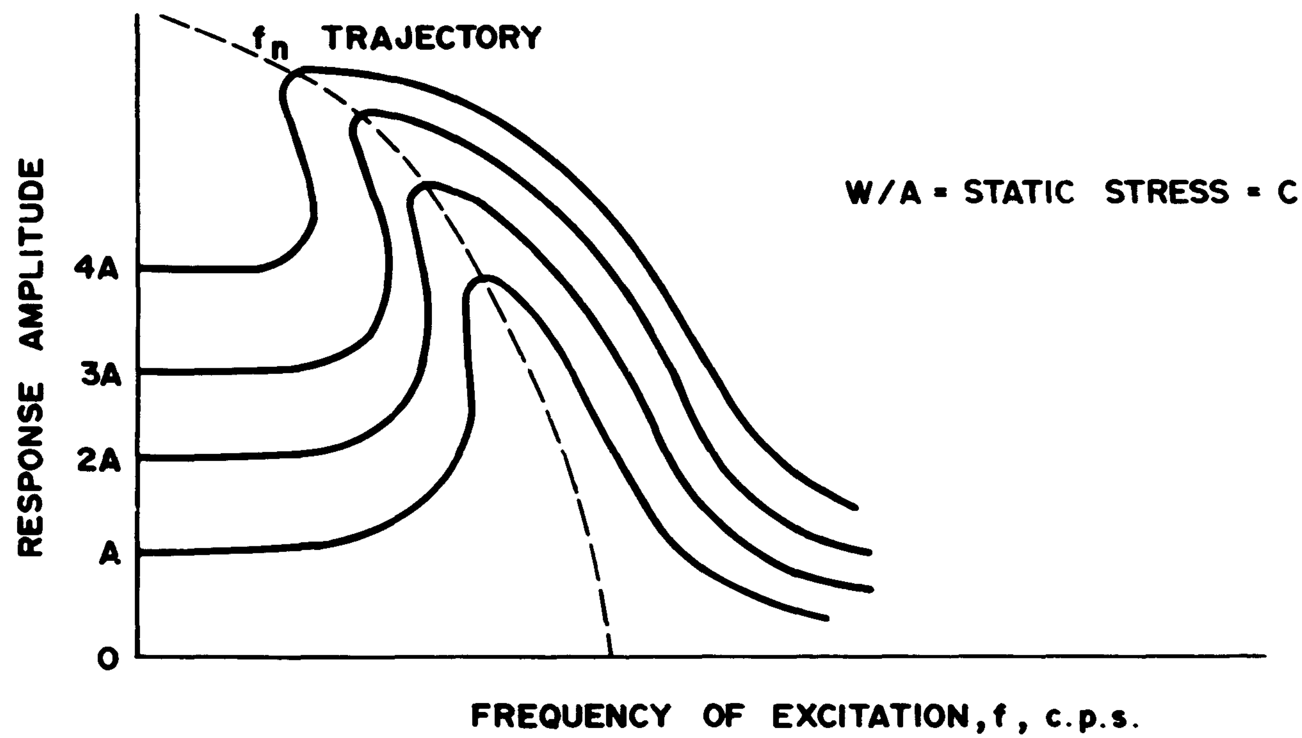

Fig 19 Response of nonlinear softening cushioning, showing effect of excitation amplitude

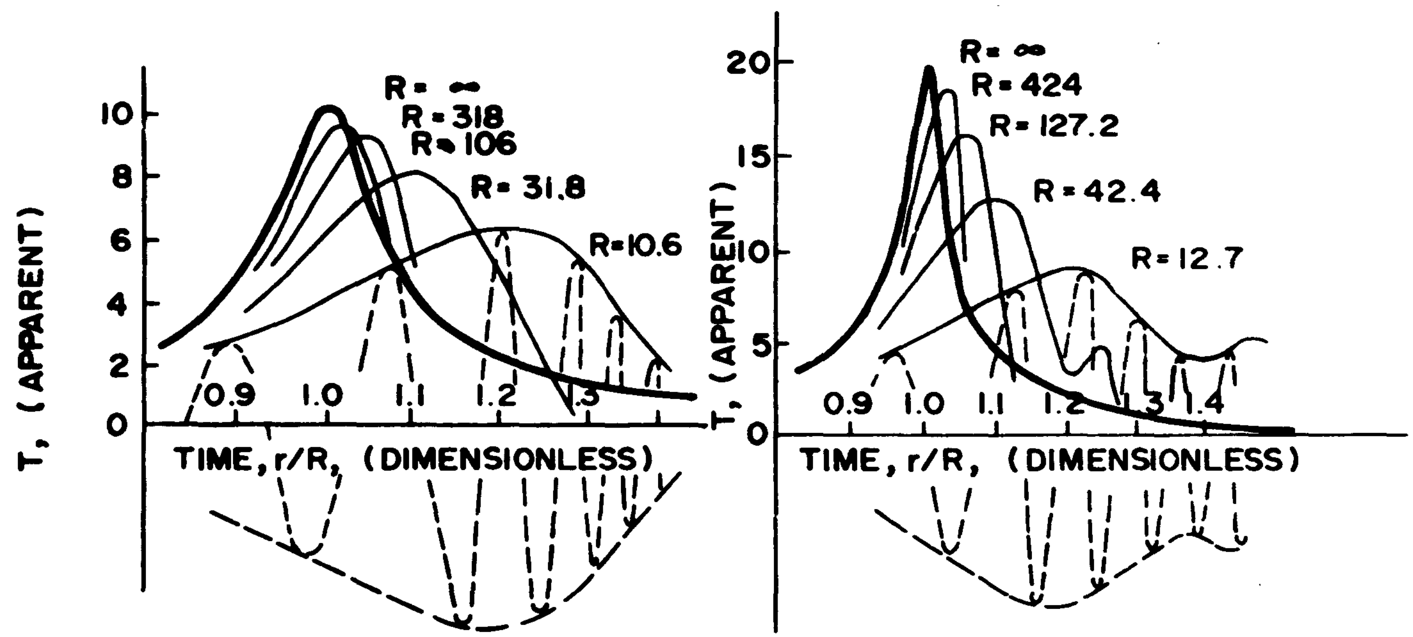

Fig 20 Effect of sweep rate on indicated transmissibility for single-degree-of-freedom system

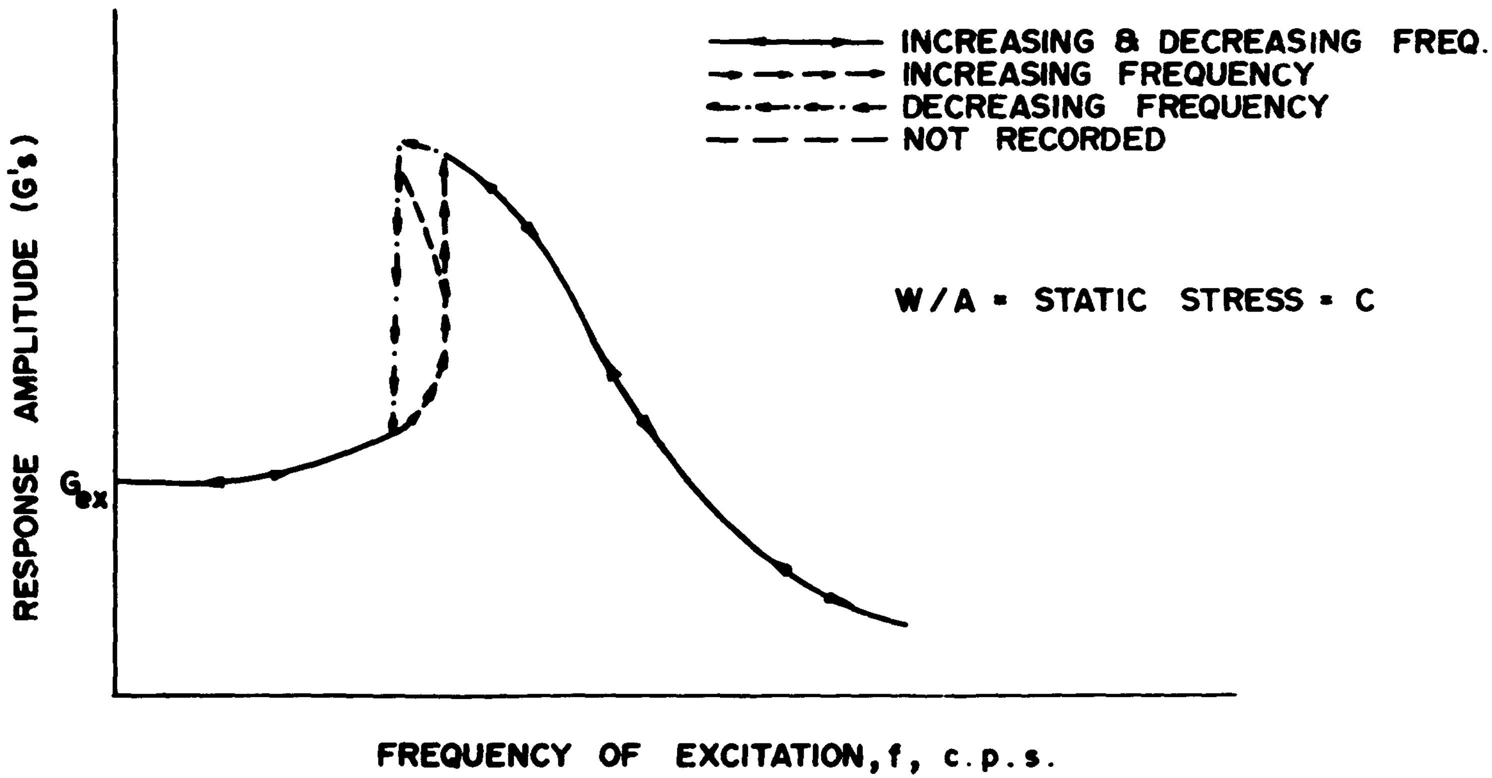

Fig 21 Response "jumps" in sweep frequency test of softening cushion

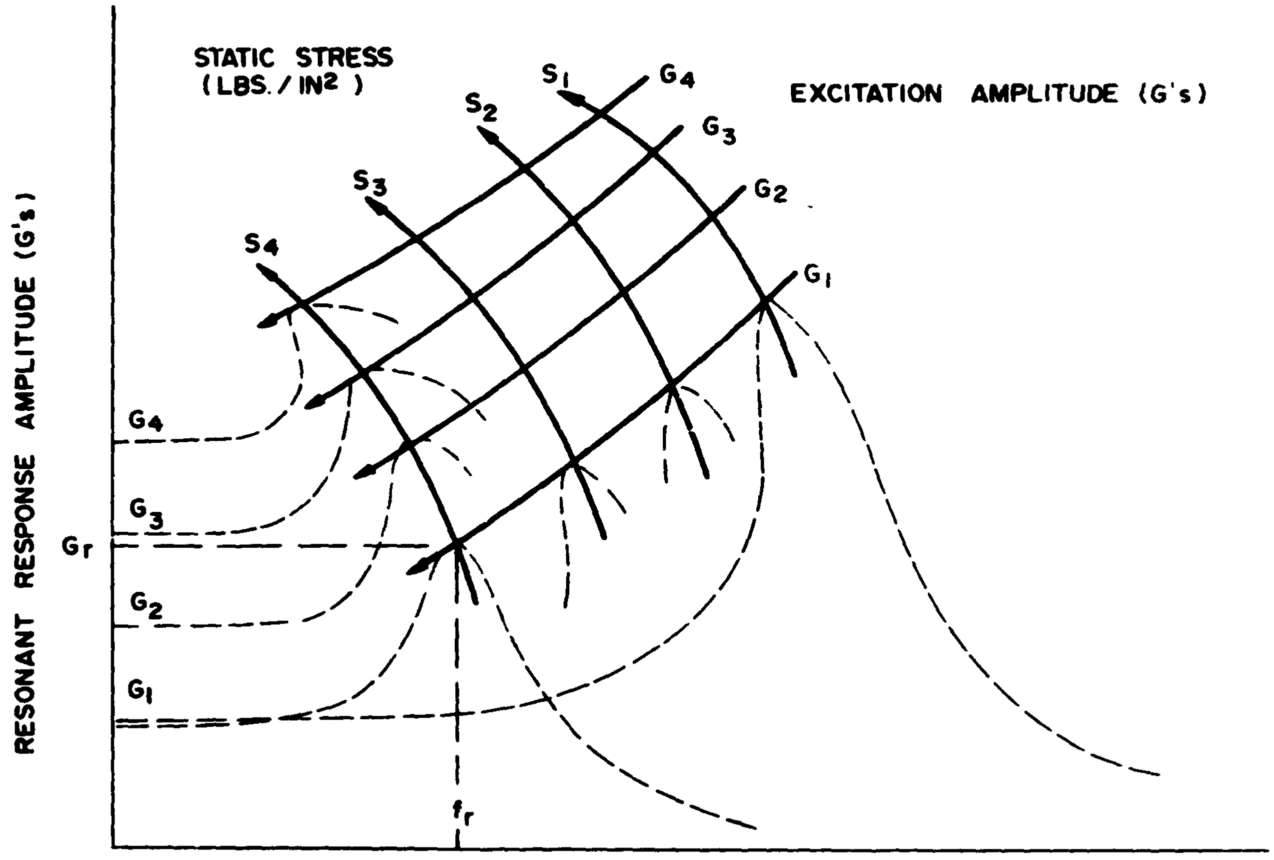

Fig 22 Resonant response (magnitude and frequency) vs input level and static stress (cushion material "A", thickness "T")

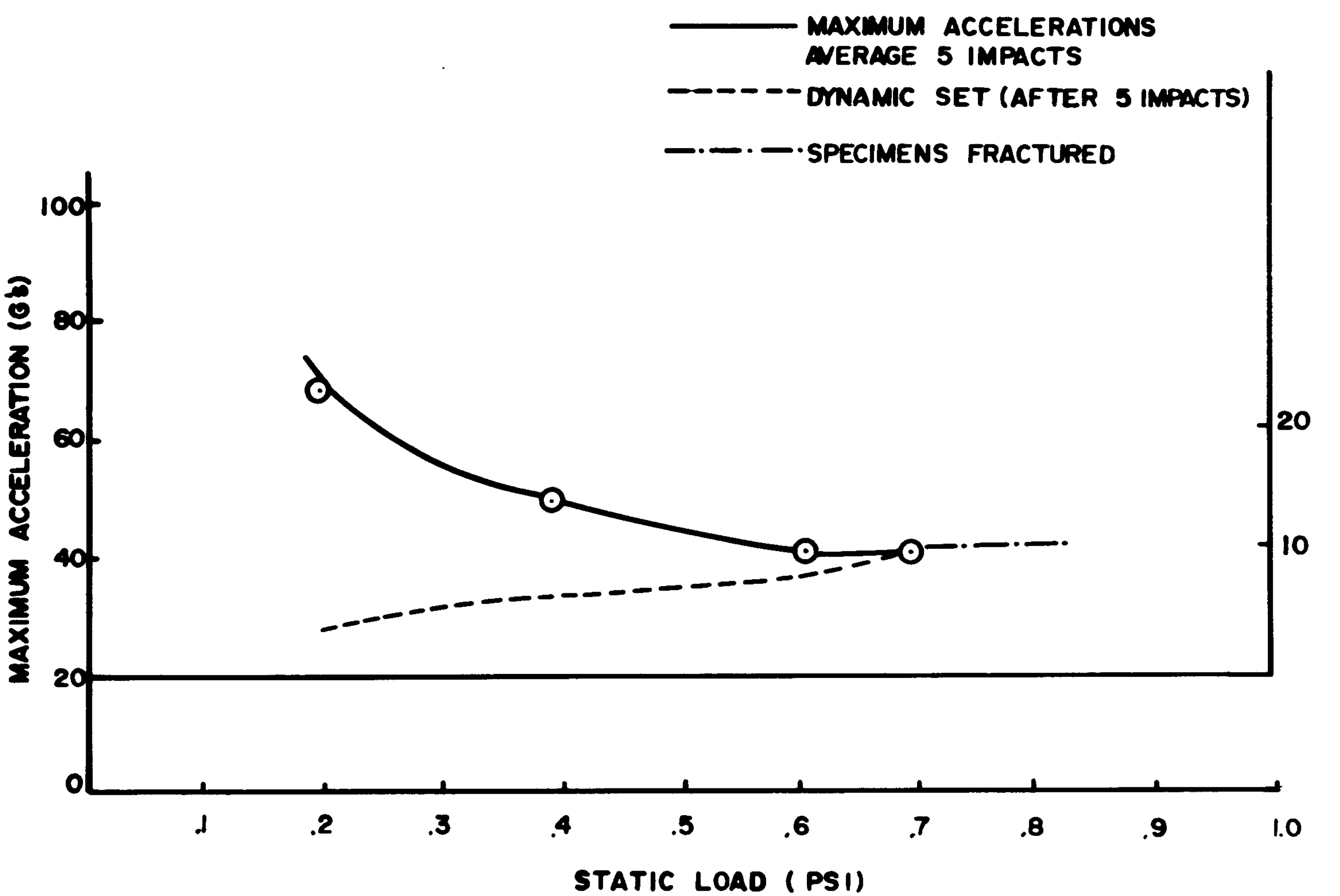

Fig 23 Maximum acceleration and dynamic set for 5-inch-thick, 0.5-lb-per-ft^3 resilient polystyrene foam. Average of 5 impacts

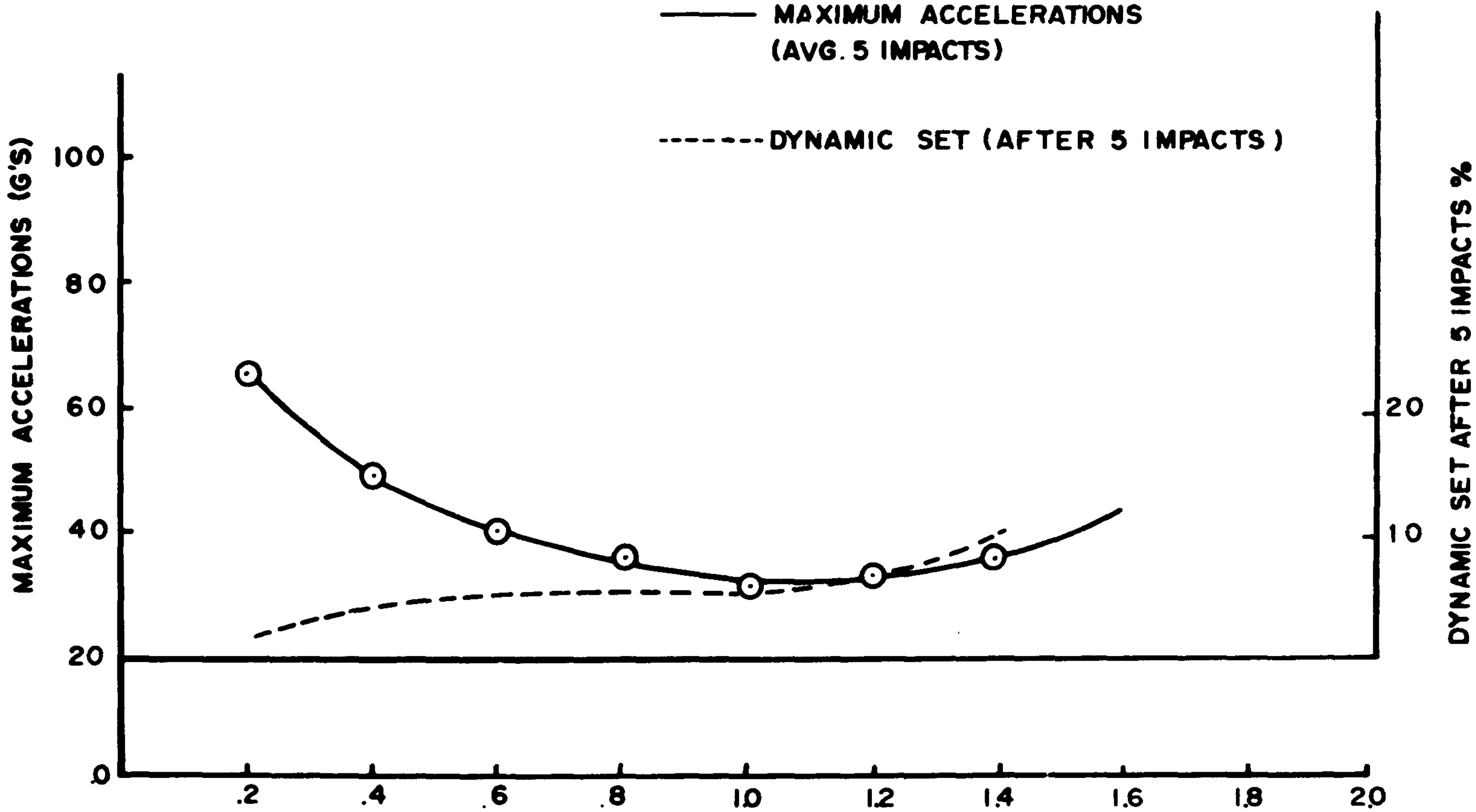

Fig 24 Maximum acceleration and dynamic set for 5-inch-thick, 0.7 lb-per-ft³ resilient polystyrene foam. Average of 5 impacts

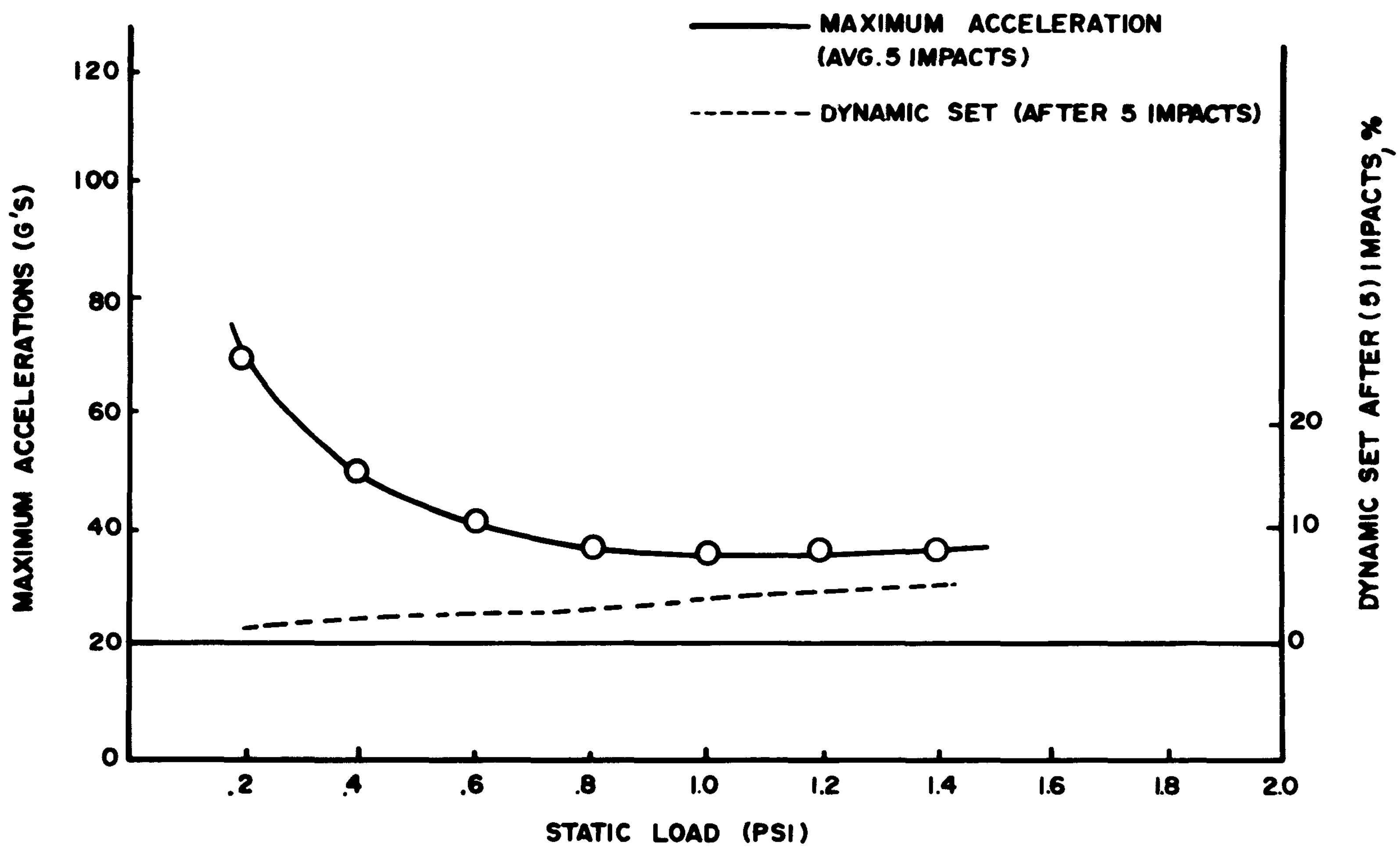

Fig 25 Maximum accelerations and dynamic set for 5-inch-thick, 1.5 lb-per-ft^3 resilient polystyrene foam. Average of 5 impacts

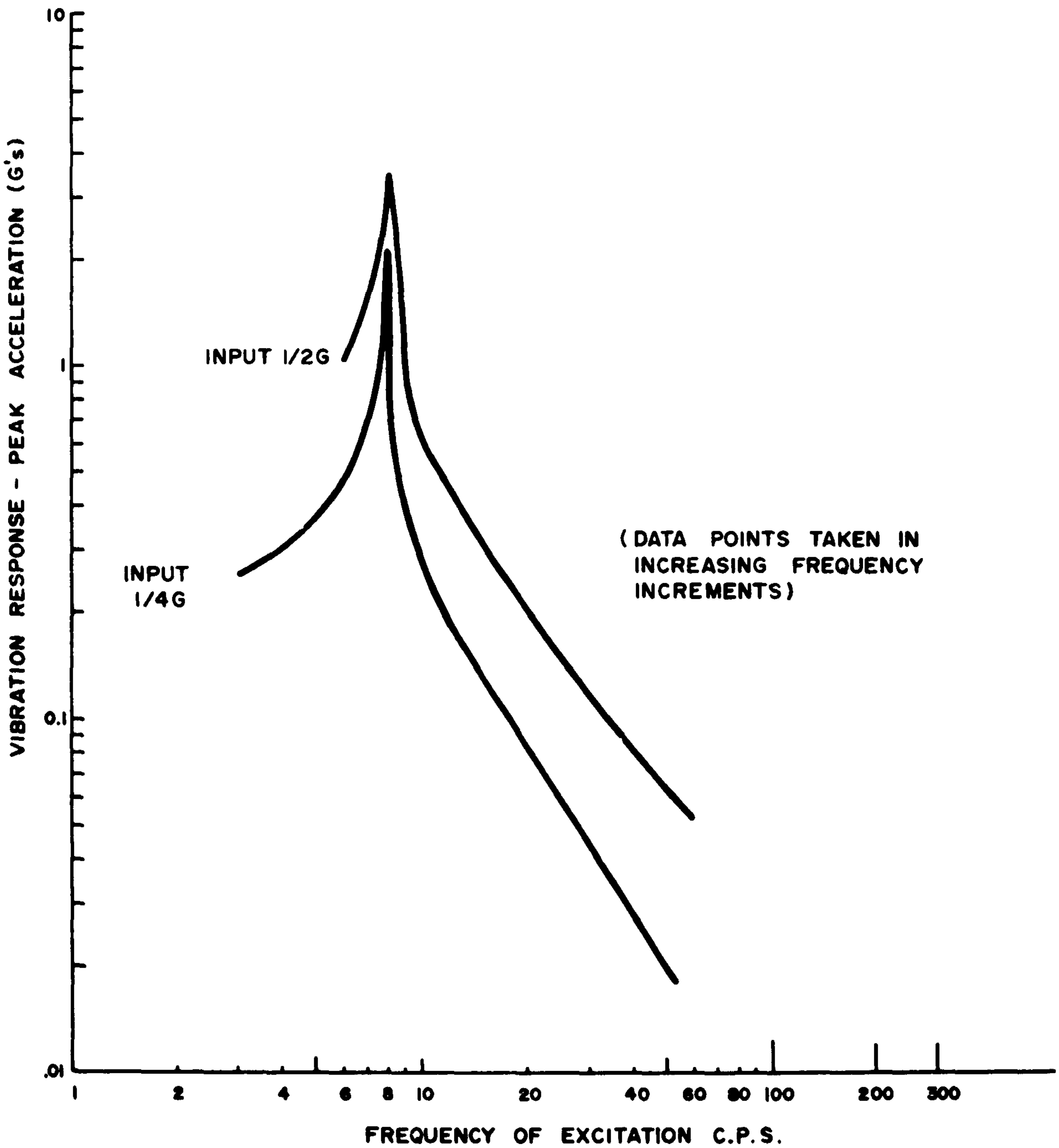

Fig 26 Vibration response vs frequency for 5-inch-thick resilient polystyrene foam. Nominal density: 0.8 lb/ft^3. Static stress: 1.46 psi

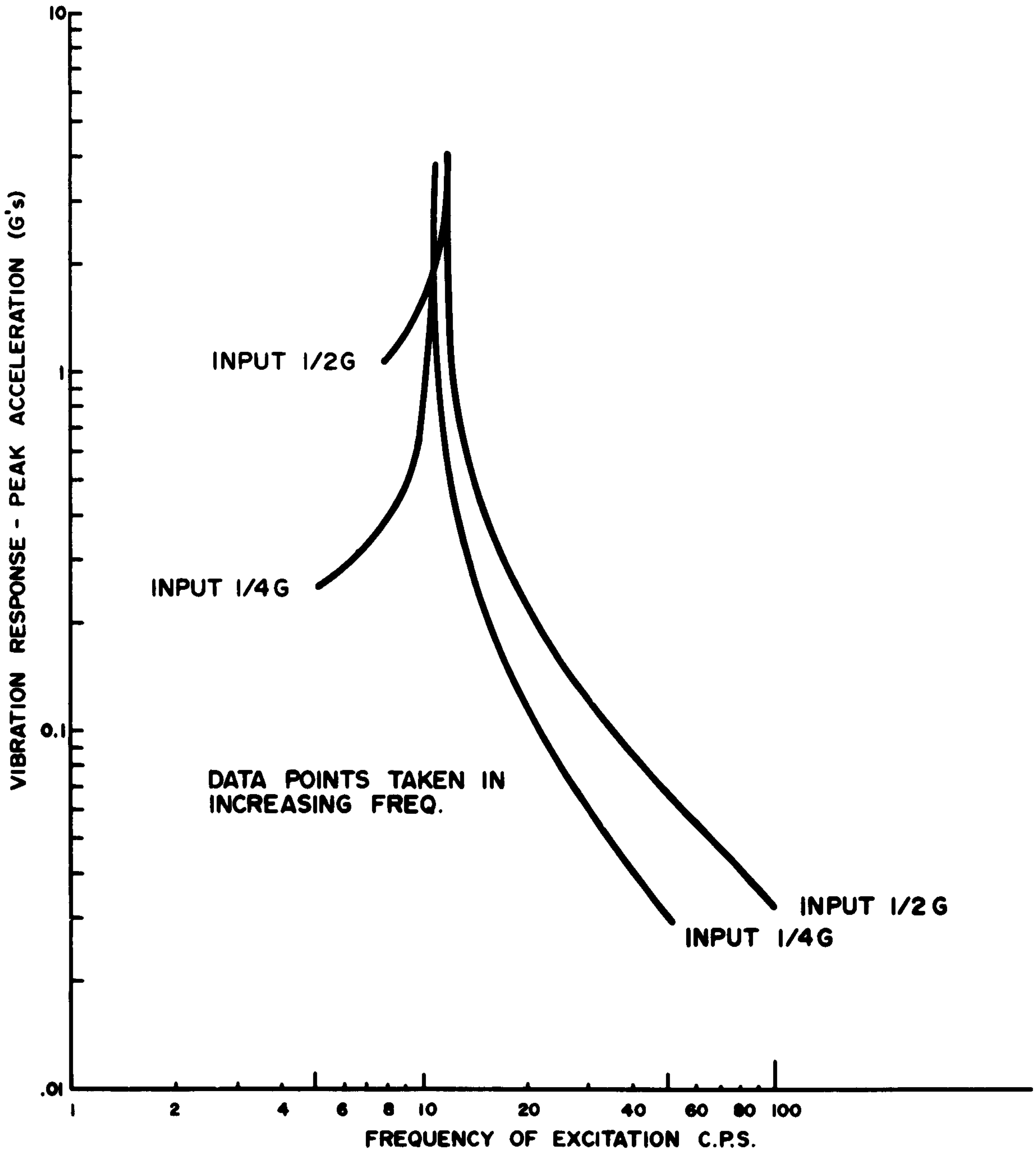

Fig 27 Vibration response vs frequency for 5-inch-thick resilient polystyrene foam. Nominal density: 0.8 lb/ft^3. Static stress: 0.71 psi

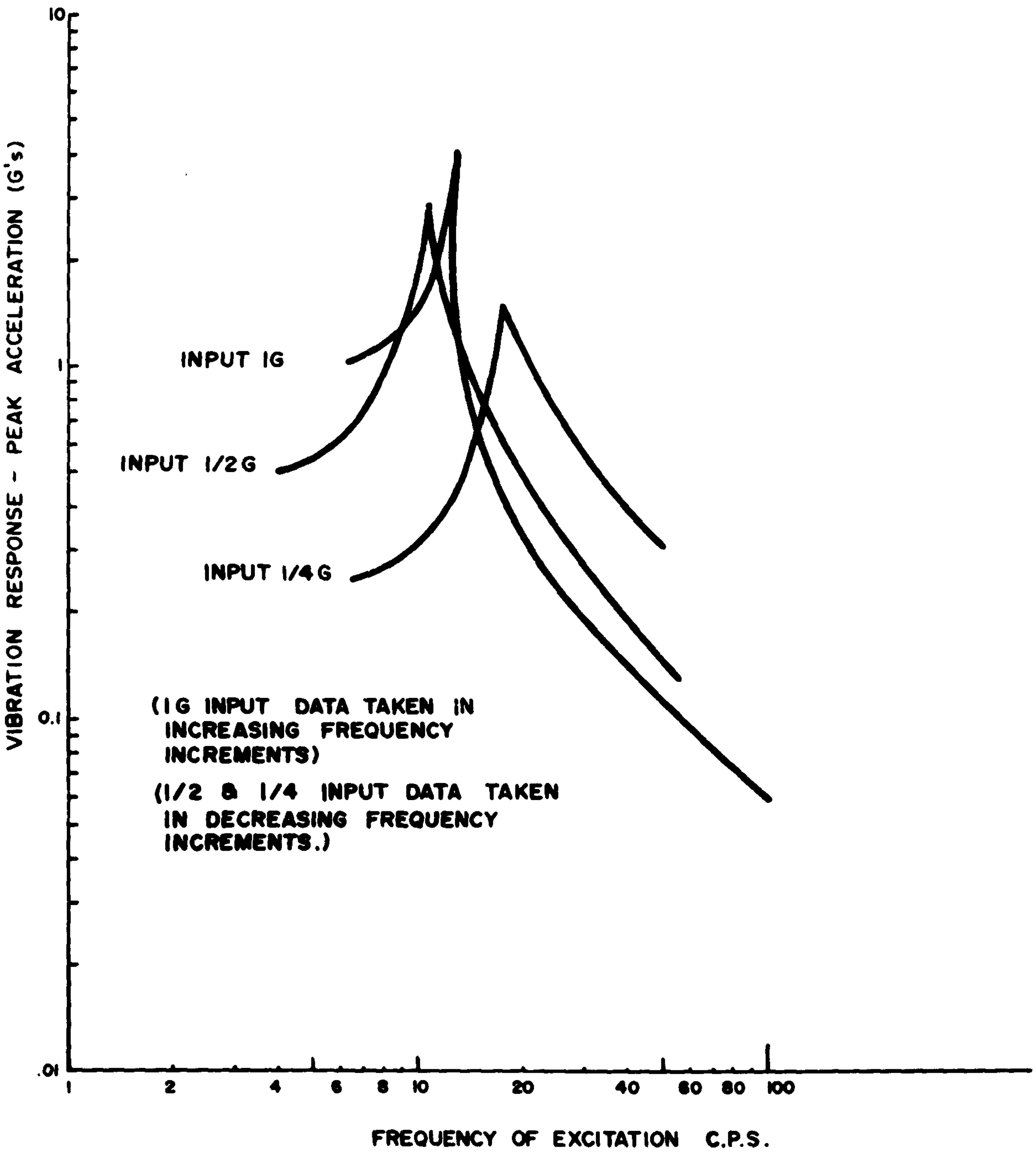

Fig 28 Vibration response vs frequency for 5-inch-thick resilient polystyrene foam. Nominal density: 0.5 lb/ft^3. Static stress: 0.6 psi

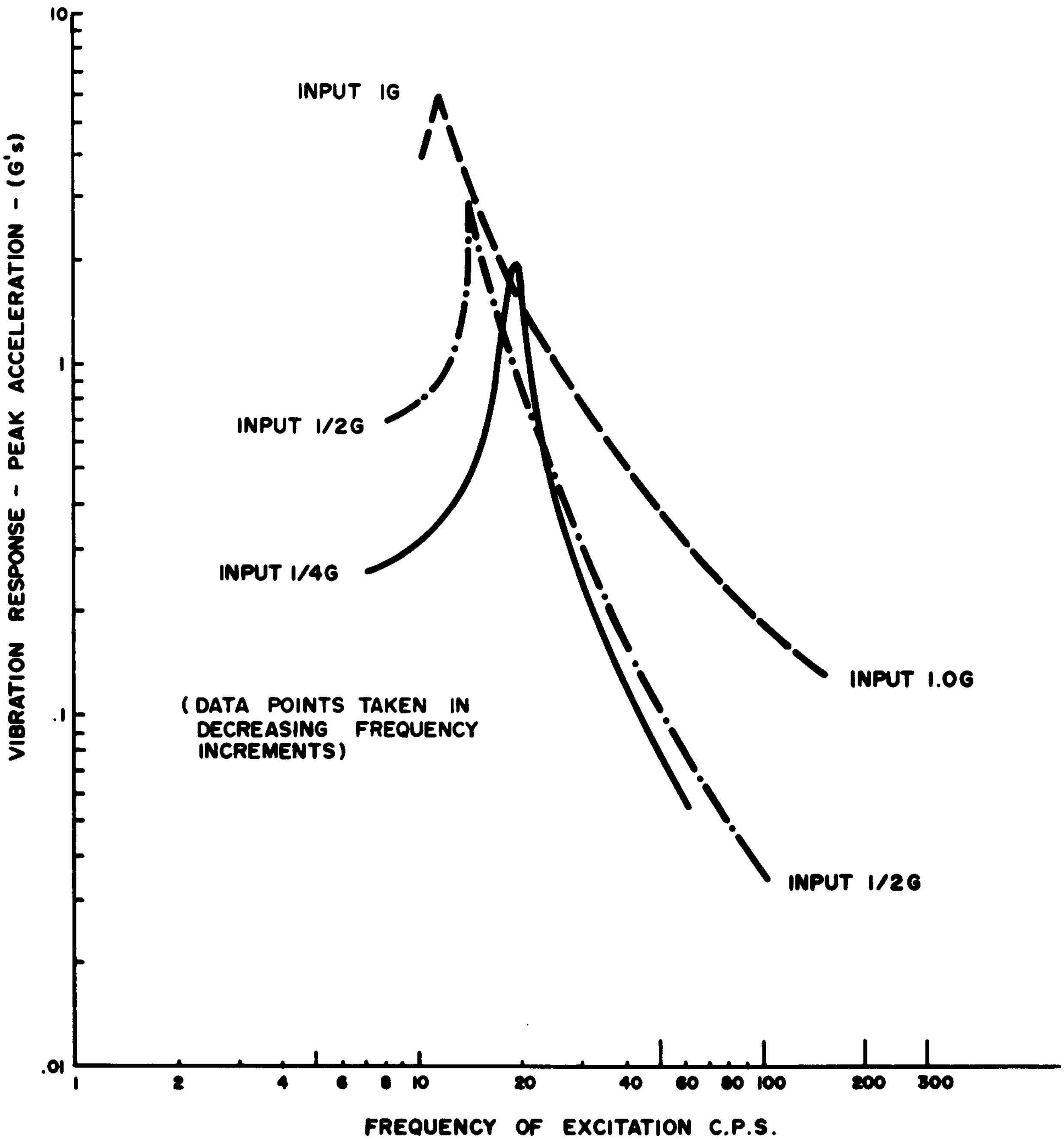

Fig 29 Vibration response vs frequency for 5-inch-thick resilient polystyrene foam. Density: 1.1 lb/ft^3. Static stress: 0.6 psi

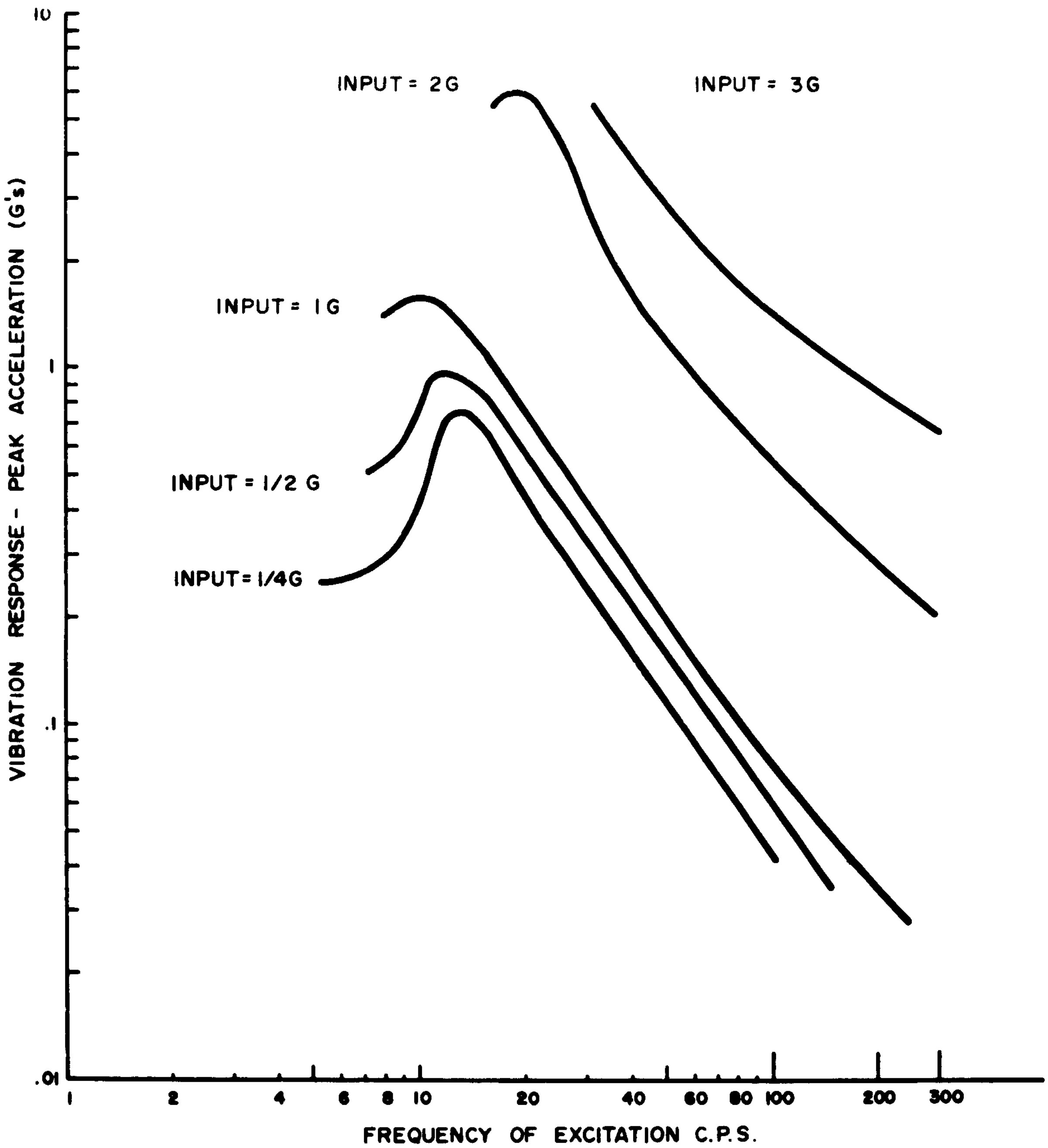

Fig 30 Vibration response vs frequency for 2.9-inch-thick resilient polyether urethane foam. Density: 2 lb/ft^3. Static stress: .16 psi

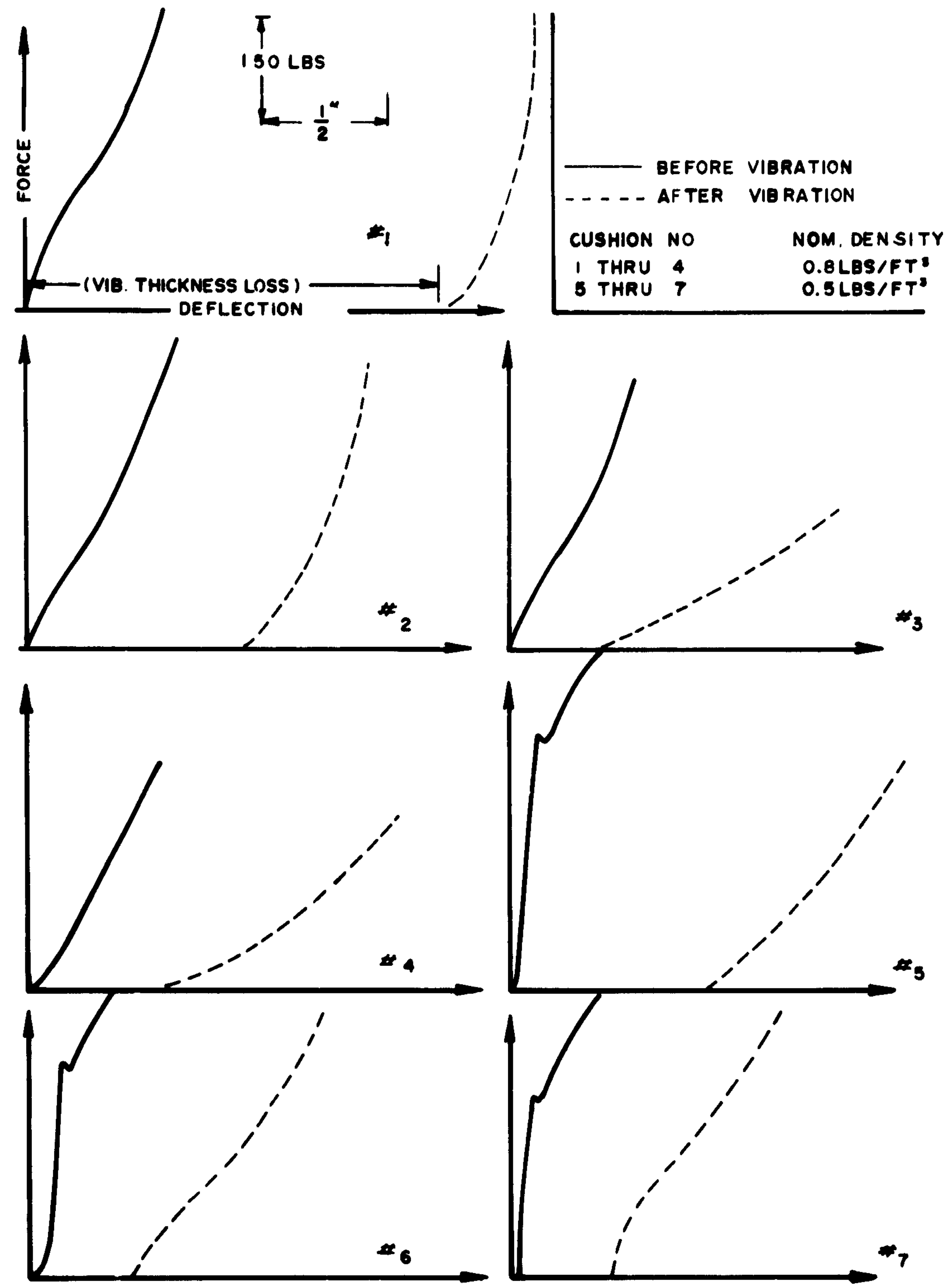

Fig 31 Force vs deflection curves for 5-inch-thick resilient polystyrene foam. Nominal densities of 0.8 and 0.5 lb/ft³

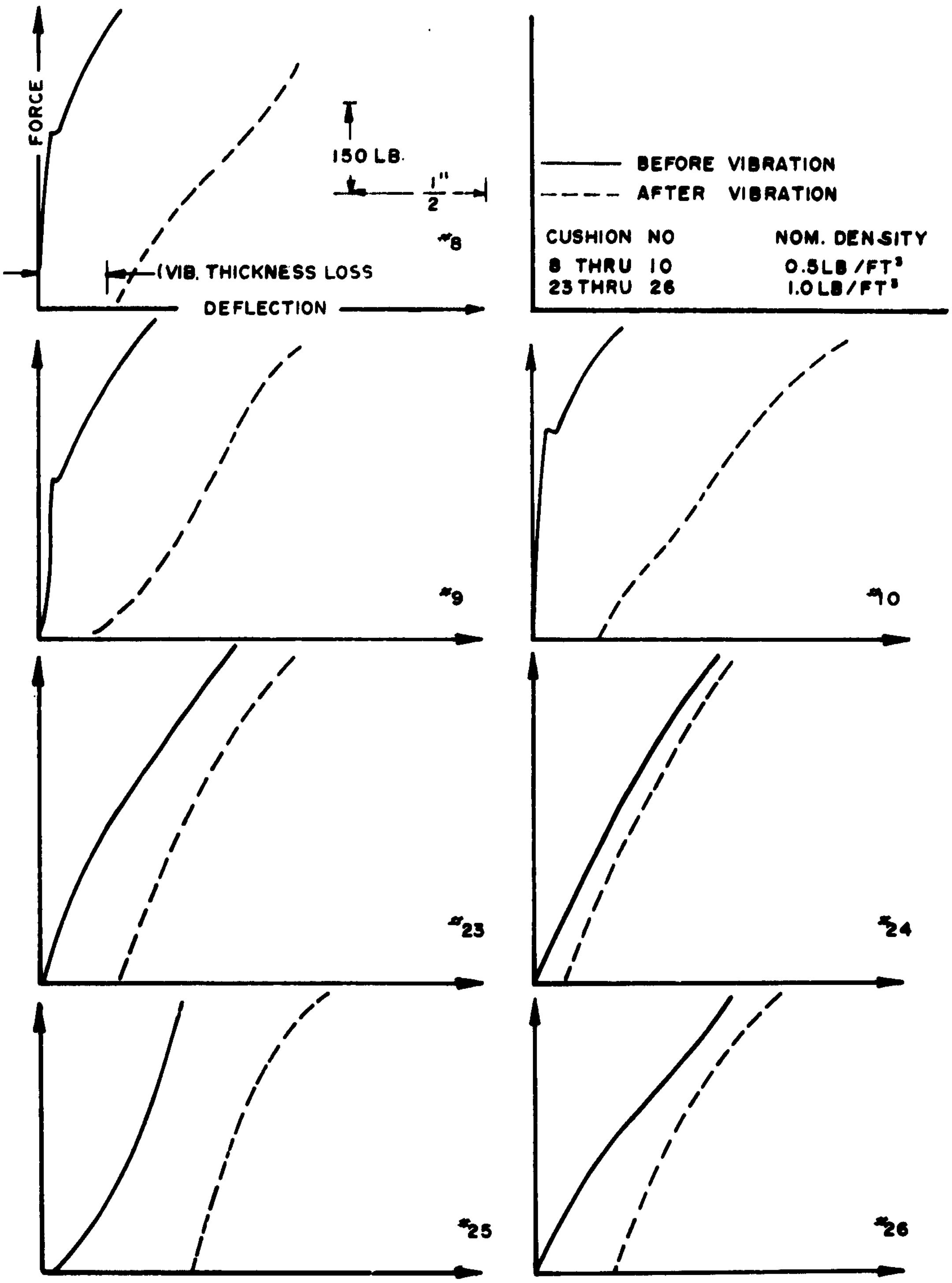

Fig 32 Force vs deflection curves for 5-inch-thick resilient polystyrene foam. Nominal densities of 0.5 and 1.0 lb/ft³

APPENDIX A

Proposed Standard Method of Test to Determine the Vibration Transmission Characteristics of Package Cushioning Materials

Scope

1. This procedure defines a method of test for use when determining vibration transmission characteristics of a cushioning material in two test configurations. When the testing has been accomplished according to the test procedure, the response of a specific material to a particular input will be known. This method of test utilizes simple harmonic motion.

Significance of Test

2. Vibration transmissibility for a package cushioning material can be determined by this test. Transmissibility for other similarly tested materials can be compared; accordingly, an aid is furnished to the package designer for selecting appropriate cushioning materials.

Definitions

3. (a) Cushioning Material – Any material used as a shock and/or vibration isolator, normally interlocking or bonded fibers, or elastomeric substances.

(b) Test Configuration – Either of two arrangements used in this method to test cushioning materials.

(c) Simple Harmonic Motion – A motion such that the displacement is a sinusoidal function of time.

(d) Amplitude – The maximum value of a sinusoidal quantity.

(e) Double Amplitude – The peak-to-peak value of amplitude.

(f) Period – The smallest increment of time for which the simple harmonic motion repeats itself.

(g) Frequency – The frequency of simple harmonic motion is the reciprocal of the period.

(h) Acceleration – A vector quantity that specifies the time rate of change of velocity.

(i) Transmissibility – The nondimensional ratio of the response amplitude of a system in steady-state forced vibration to the excitation amplitude. The ratio may be one of forces, displacements, velocities, or accelerations.

(j) Resonant Frequency – The frequency at which resonance occurs.

(k) Resonance – Resonance of a system in forced vibration exists when any change, however small, in the frequency causes a decrease in the response of the system.

(l) Q (Quality Factor) – The quantity Q is a measure of the sharpness of resonance or frequency selectivity of a resonant vibratory system having a single degree of freedom.

(m) Bearing Stress – The static compressive stress developed as a result of a load placed on a particular area, expressed in pounds per square inch.

(n) Strain – The change in length per unit length resulting from the application of a static, axial, uniform bearing stress, expressed in inches per inch.

(o) Vibration Machine – A device for subjecting a mechanical system to controlled and reproducible mechanical vibration.

(p) Vibration Fixture – A rigid structure affixed to a vibration machine and used to limit the motion of the test material.

Apparatus

4. (a) Vibration Machine capable of producing simple harmonic motion with a wave form distortion less than 3 per cent.

(b) Instrumentation by which the vibration input and response can be measured. The measuring system shall have an accuracy of + 5 per cent.

(c) A vibration fixture similar to Figure 1.[1] The fixture and load shall conform to the following requirements:

(1) The base, top plate, and load shall have length and width dimensions which are greater than the corresponding length and width dimensions of the specimen. Vertical motion of the load shall not be constrained by the fixture.

(2) The lowest resonant frequency of the fixture and load shall be above 500 cycles per second.

Test Configuration Description

5. Two types of test configuration which may be used are:

(a) Compression configuration – Two identical specimens are placed in the fixture in the following order: one test specimen is placed on the fixture base, followed by the load, the top specimen, and the top plate secured in place to the fixture.

(b) Compression-Tension Configuration – The test specimen is firmly attached between the base and load.

Test Specimen Description

6. Test specimens shall be right square prisms of the largest size practical. The minimum bearing area shall be four inches by four inches.

Number of Specimens

7. The number of specimens shall be selected according to statistical sampling plans such as:

(a) Recommended Practice for Probability Sampling of Materials (ASTM E-105-58).

---

[1]Figures 1, 2, and 3 of this appendix were not reproducible. The information in these figures may be found in the report as indicated below.

Figure 1: The fixture shown is similar to Figure 3 of the report.

Figure 2: The sinusoidal input prescribed here is the heavy curve in Figure 10 of the report.

Figure 3: The sweep rate specified is discussed on pages 20 and 21.

(b) Recommended Practice for Choice of Sample Size to Estimate the Average Quality of a Lot or Process (ASTM E-122-58).

Test Procedure

(a) Preworking – Preworking shall be optional.

(b) Specimen Size – The length and width shall be measured to the nearest 0.01 inch.

(c) Specimen Thickness – The top surface shall be uniformly loaded to a bearing stress of 0.025 pounds per square inch. After a 30-second interval, and while the specimen is still under load, the thickness shall be measured to the nearest 0.01 inch. The measurement shall be made at the geometric center of the top surface of the specimen. An alternate method of measuring thickness is to average the thickness of the four corners of the specimen. Record this value as the original thickness, t_o.

(d) Specimen Weight – Specimen weight shall be measured to an accuracy of one percent.

(e) Bearing Stress – Five different values of bearing stress shall be used.

(f) Mounting the Specimen – The specimen shall be centered in the test fixture. When using test configuration 5(a), the top plate shall be secured to the fixture so that the top specimen will be subjected to a strain of one percent.

(g) Dynamic Input – The specimen shall be subjected to a sinusoidal input as described in Figure 2. Other constant acceleration inputs such as $\pm 1g$, $\pm 2g$, and $\pm 5g$ are recommended whenever possible. The frequency sweep rate which shall be used is shown in Figure 3. This will insure that maximum transmissibility values of 95 percent of resonance during steady state vibration are attained.

(h) Final Specimen Thickness – Within five minutes after the vibration test for any given specimen is completed, the specimen thickness shall be measured according to section 8(c). This value shall be recorded as the final thickness, t_f

Calculations

9. (a) Density – The density shall be calculated:

$$D = \frac{W}{L_1 \times L_2 \times t_o}$$

where

D = density in pounds per cubic foot

W = specimen weight, in pounds

L_1 = specimen length, in feet

L_2 = specimen width, in feet

t_o = original specimen thickness in feet.

(b) Bearing stress – The bearing stress for each specimen size and load shall be calculated:

$$S_b = P/A$$

where

S_b = bearing stress, in pounds per square inch

P = load, in pounds

A = area of contact, in square inches.

(c) Transmissibility – The transmissibility for a selected frequency shall be calculated:

$$T = a_r/a_i$$

where

T = transmissibility, dimensionless

a_r = measured peak acceleration response at the frequency selected, a multiple of g

a_i = measured input peak acceleration input at the frequency selected, a multiple of g.

(d) Thickness loss – The thickness loss shall be calculated:

$$\text{Percent thickness loss} = \frac{(t_o - t_f) \times 100}{t_o}$$

where

t_o = original cushion thickness, in inches

t_f = final cushion thickness, in inches.

Report

10. (a) A description of the material, including the name of the manufacturer, the manufacturer's designation, the specification compliance, and the date tested.

(b) The original dimensions of each test specimen as measured in 8(b) and 8(c).

(c) The density of each test specimen as calculated in 9(a).

(d) The bearing stress to which each specimen was subjected, as calculated in 9(b).

(e) A plot of acceleration input and response vs the input frequency for each combination of bearing stress, input, sweep rate, and thickness of the specimen tested.

(f) The sweep rate to which the specimens were tested.

(g) The thickness loss that each test specimen experienced as calculated in 9(d).

(h) The temperature and humidity at which the test was conducted.

(i) A detailed explanation of any deviations from this method of test.

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